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This Month's Cover: Cross section and phantom view of a Hoover cleaner. Typical of the high development of modern tools for the housewife, this cleaner has a built-in light, vertical motor and fan, belt-driven agitator, and wheel adjustment for cleaning conditions.

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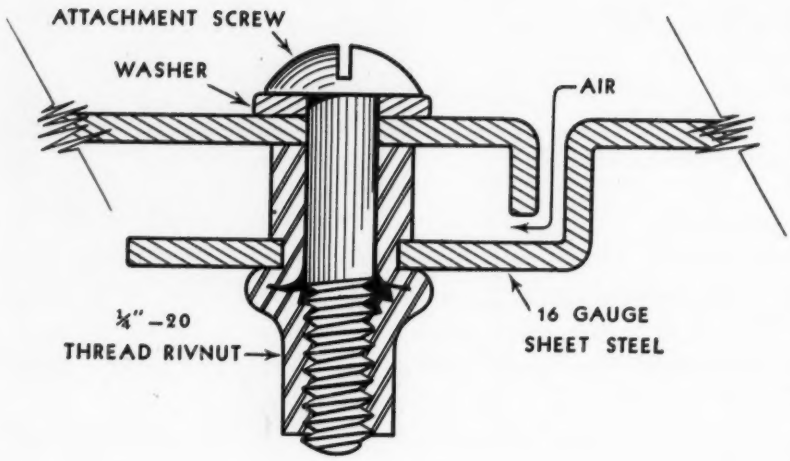
Acceptance under Act of June 5, 1934. Authorized July 20, 1934

New ¼" RIVNUT solves tough fastening problem—replaces three parts



Designers of a new bus engine came up against a tough fastening problem in providing an air vent between two steel sheets of the engine housing. Because it was a semi-blind fastening job, the use of conventional bolts, nuts and spacers would slow production.

They found the answer in spacer-head Rivnuts. The ¼" steel Rivnut was inserted in the 16-gauge sheet steel and upset with power tool by *one* man, working from *one* side. The increased Rivnut head served as a spacer for the second steel sheet, which was firmly anchored by an attachment screw, threaded into the Rivnut.

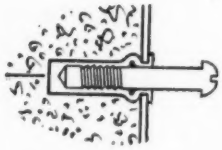
The ¼"-20 thread Rivnut provided extra strength, improved the design and cut costs. If you're faced with a fastening problem, why not call in a Rivnut engineer? Write the B. F. Goodrich Company, Dept. MD-128, Akron, O.

What's your toughest fastening problem?

TORQUE?

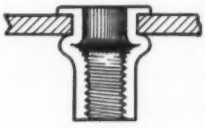
Special Rivnuts with splined shafts or key (above) supply needed torque resistance.

SEAL?



Firm grip of upset Rivnut on material, plus special closed end, gives tight seal.

MATERIAL?

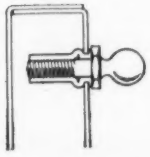


Even in thin materials, Rivnuts provide a secure nut plate with eight clean threads.

ATTACHMENT?

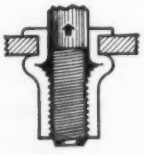
Rivnut serves both as blind rivet and nut plate, makes possible multiple fastening.

FINISH?



Rivnuts can be installed *after* finish, such as paint or porcelain, is applied.

LOADS?



Typical Rivnut thread strength: 6-32—784 lbs.; 8-32—1101 lbs.; 10-32—1220 lbs.*

RIVNUTS *lick them all!*

**Figures given for aluminum. Loads for brass and steel Rivnuts about double.*

B.F.Goodrich

RIVNUTS

It's a rivet—It's a nutplate

GET YOUR COPY OF NEW RIVNUT DATA BOOK

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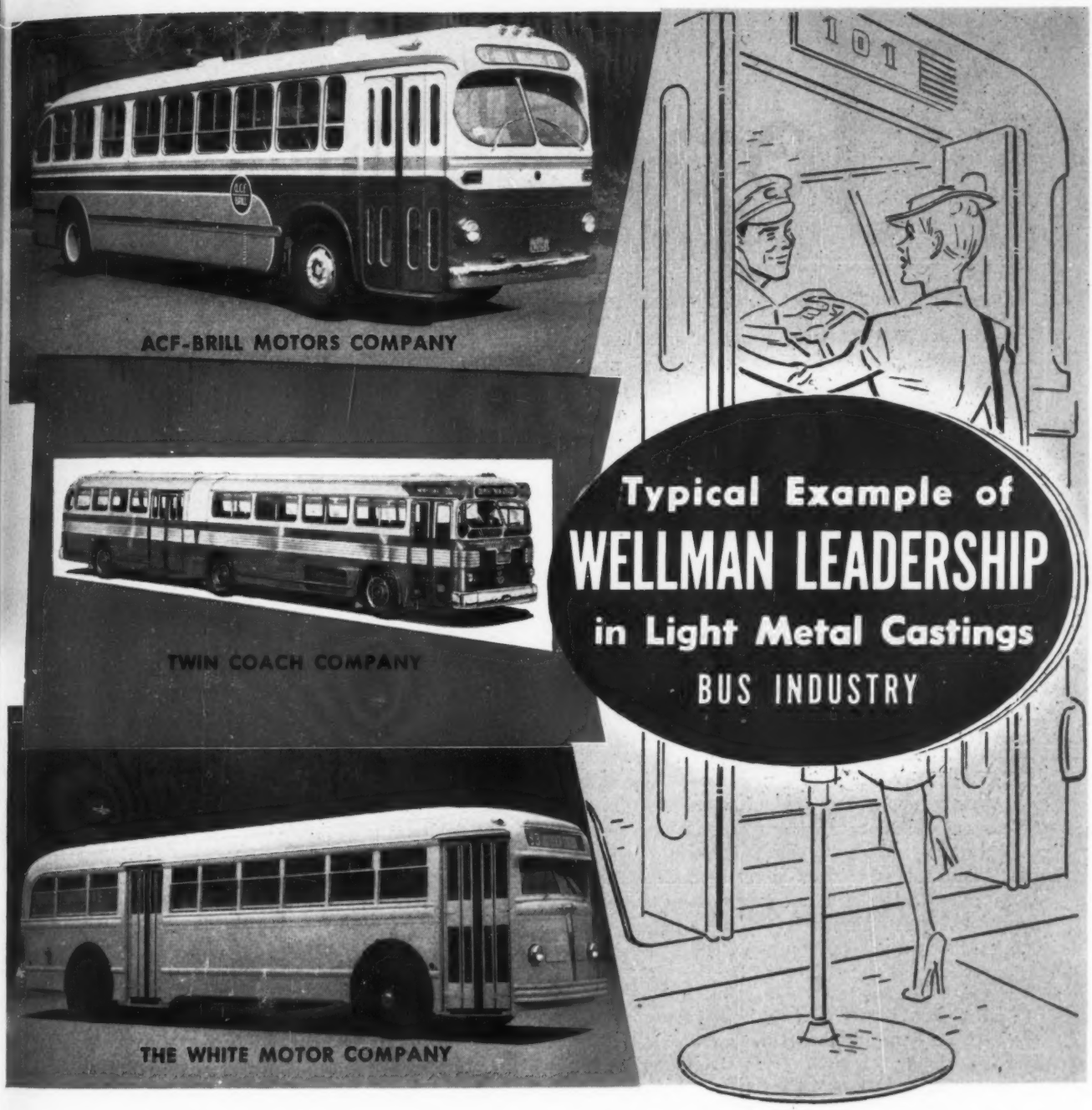

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Topics

DIFFERENCE between practice and theory, according to "Boss" Kettering of General Motors, is that in practice one cannot "forget" anything. In theory it is possible to conveniently neglect factors which would cause failure in practice.

SUPER-ALLOY centrifugal casting process, developed by Firth-Vickers Stainless Steels Ltd., of Sheffield, England to meet needs for high-strength, high-temperature service, has been licensed to Lebanon Steel Foundry. Metal is cast under pressure in centrifugal molds, producing a symmetrical ring of high density with excellent structural properties.

EXTENDED LIFE for fluorescent lamps, by eliminating or greatly reducing the formation of discoloration on the lamp tube, is the result of developments at Sylvania Electric Products Inc. Addition of a very small amount of antimony trioxide to the fluorescent coating material shows a higher efficiency at 6000 hours than other lamps after 3000 hours of use.

CERAMIC COATINGS on molybdenum show promise for high-temperature applications. In air atmosphere at 1650 F the coated material showed no deterioration during a 70-hour test. The coating also provides short-time protection at 2600 F, making the material feasible for special high-temperature applications in which prolonged service is not required.

PETROLEUM DEPOSITS may be discovered by a more economical method than formerly. Developed by Thomas C. Poulter, the method employs a pattern of small, specially shaped charges of explosive composition which are detonated above the ground as contrasted with conventional methods of firing a single, large charge in a shot hole at various depths beneath the surface.

FIBERGLAS CLOTH is being used as a series cutout in Westinghouse lamps. These 30-volt lamps operating in series on 600-volt dc circuits have glass cloth inserted between the terminal wires so that, when one burns out, the momentary 600-volt potential impressed across the wires breaks down the insulation and fuses the wires. This keeps the remaining lamps in service until the burned-out lamp is replaced.

COPPER-BASE alloy, known as Fansteel 607 metal and developed by Fansteel Metallurgical Corp., is useful for current-carrying parts which are subject to bending, fatigue, abrasion, heat and impact. When heat treated, the alloy will maintain constant spring tension better than common spring materials.

SEVEN NEW standard samples for rubber compounding are available from the National Bureau of Standards. Government specifications are now predicated on the use of these samples for stress-strain testing of synthetic rubbers.

BRAZING magnesium is now a satisfactory process, according to The Dow Chemical Co. A flux must be present with the new method to free surfaces from oxide or other films, permitting capillary flow.

TWO-TONE electroplating of gold and chromium by utilizing copper-nickel masks has been employed by the Continental Die Casting Corp. for producing the hood ornament on 1949 Cadillacs.

TRACK TESTS have begun on the first gas turbine-electric locomotive to be built and operated in the United States. The 4500-hp locomotive, built by the American Locomotive and General Electric companies will be placed in operation on a demonstration basis next spring by the Union Pacific Railroad. Present fuel is bunker "C" oil but it is hoped that coal may be used in future designs.

PHOTOELECTRIC control is built into new GE street lighting units. Designed for use on multiple circuits each luminaire is adjustable to operate between 1 and 6 foot-candles. Electronic unit is housed in a glass hood at top of the unit.

MACHINE DESIGN

Rocket Motors

. . . practical design problems and their solution

By George P. Sutton
Supervisor, Propulsion Development
North American Aviation, Inc.
Los Angeles

RIGOROUS operating conditions of rocket motors pose engineering problems which make stringent demands on the designer. He has to allow for high heat transfer across stressed walls, establish the level of precision of fabrication tolerances, provide for high pressures, and keep the weight of the unit to a practical minimum. This article illustrates several of these design problems and indicates some solutions. Discussion is confined to liquid-propellant rocket motors, which can withstand repeated and prolonged firing.

Thrust of a rocket unit is derived from the momentum of ejected hot gases. Rocket propellants, i.e., a fuel and a liquid oxidizing chemical (called an oxidizer) are burned at high pressure in a combustion chamber and the resulting product gases are accelerated to a high velocity in a supersonic nozzle. Flame temperatures are on the order of 5000 F and the gas velocities obtained usually exceed one mile per second. The theoretical relations which determine the design conditions and the required size and shape of the rocket motor can be found in various texts, two of which are listed at the end of the article.

PRECISION REQUIREMENTS: *Fig.* 1 shows the basic

fastening problem?

Special Rivnuts with splined shafts or key (above) supply needed torque resistance.

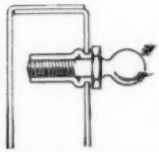
Firm grip of upset Rivnut on material, plus special closed end, gives tight seal.

Even in thin materials, Rivnuts provide a secure nut plate with eight clean threads.

ATTACHMENT?

Rivnut serves both as blind rivet and nut plate, makes possible multiple fastening.

FINISH?



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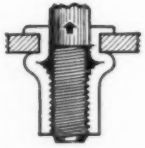

Typical Rivnut thread strength: 6-32—784 lbs.; 8-32—1101 lbs.; 10-32—1220 lbs.*

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**Figures given for aluminum. Loads for brass and steel Rivnuts about double.*

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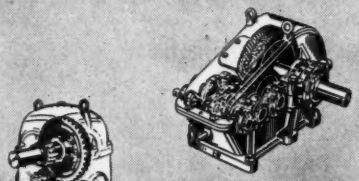
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WORM GEAR SPEED REDUCERS

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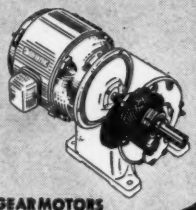
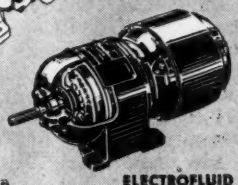
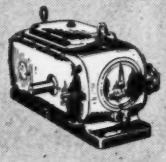

GEARMOTORS


ELECTROFLUID DRIVES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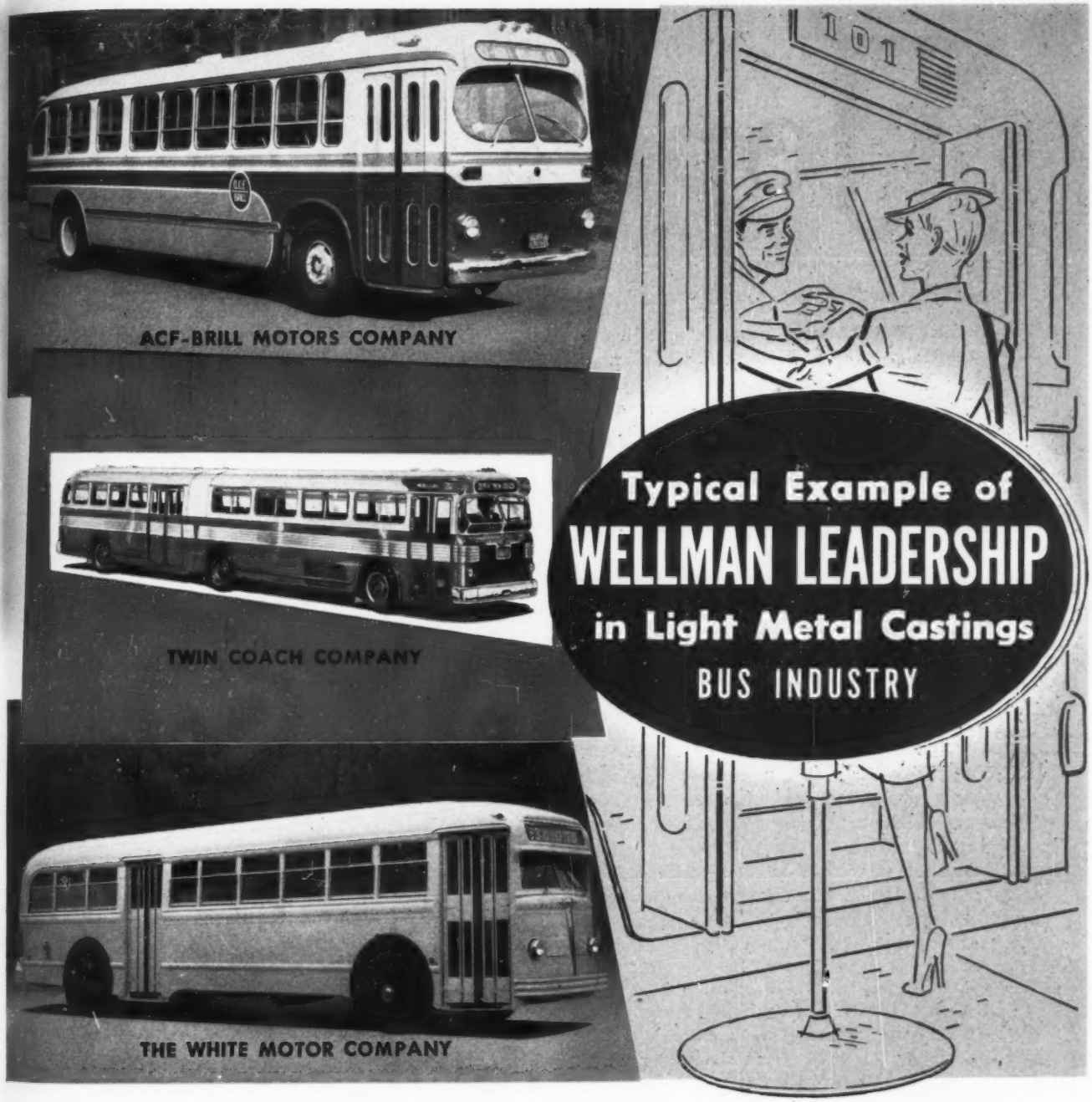

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Topics

DIFFERENCE between practice and theory, according to "Boss" Kettering of General Motors, is that in practice one cannot "forget" anything. In theory it is possible to conveniently neglect factors which would cause failure in practice.

SUPER-ALLOY centrifugal casting process, developed by Firth-Vickers Stainless Steels Ltd., of Sheffield, England to meet needs for high-strength, high-temperature service, has been licensed to Lebanon Steel Foundry. Metal is cast under pressure in centrifugal molds, producing a symmetrical ring of high density with excellent structural properties.

EXTENDED LIFE for fluorescent lamps, by eliminating or greatly reducing the formation of discoloration on the lamp tube, is the result of developments at Sylvania Electric Products Inc. Addition of a very small amount of antimony trioxide to the fluorescent coating material shows a higher efficiency at 6000 hours than other lamps after 3000 hours of use.

CERAMIC COATINGS on molybdenum show promise for high-temperature applications. In air atmosphere at 1650 F the coated material showed no deterioration during a 70-hour test. The coating also provides short-time protection at 2600 F, making the material feasible for special high-temperature applications in which prolonged service is not required.

PETROLEUM DEPOSITS may be discovered by a more economical method than formerly. Developed by Thomas C. Poulter, the method employs a pattern of small, specially shaped charges of explosive composition which are detonated above the ground as contrasted with conventional methods of firing a single, large charge in a shot hole at various depths beneath the surface.

FIBERGLAS CLOTH is being used as a series cutout in Westinghouse lamps. These 30-volt lamps operating in series on 600-volt dc circuits have glass cloth inserted between the terminal wires so that, when one burns out, the momentary 600-volt potential impressed across the wires breaks down the insulation and fuses the wires. This keeps the remaining lamps in service until the burned-out lamp is replaced.

COPPER-BASE alloy, known as Fansteel 607 metal and developed by Fansteel Metallurgical Corp., is useful for current-carrying parts which are subject to bending, fatigue, abrasion, heat and impact. When heat treated, the alloy will maintain constant spring tension better than common spring materials.

SEVEN NEW standard samples for rubber compounding are available from the National Bureau of Standards. Government specifications are now predicated on the use of these samples for stress-strain testing of synthetic rubbers.

BRAZING magnesium is now a satisfactory process, according to The Dow Chemical Co. A flux must be present with the new method to free surfaces from oxide or other films, permiting capillary flow.

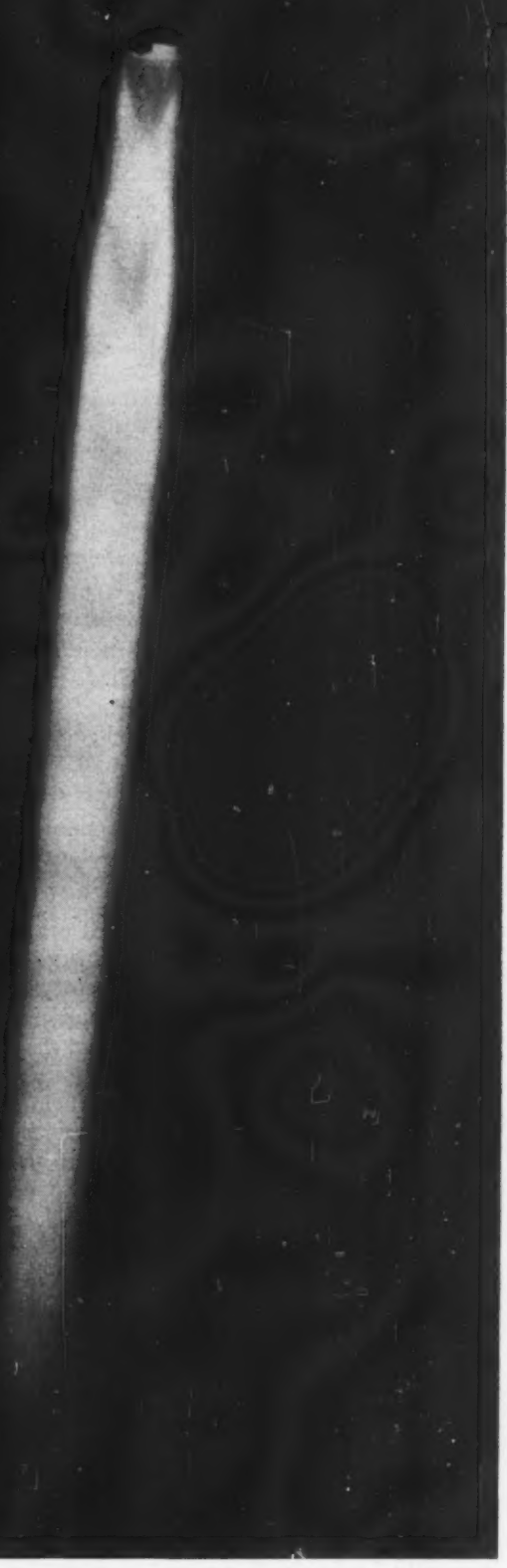
TWO-TONE electroplating of gold and chromium by utilizing copper-nickel masks has been employed by the Continental Die Casting Corp. for producing the hood ornament on 1949 Cadillacs.

TRACK TESTS have begun on the first gas turbine-electric locomotive to be built and operated in the United States. The 4500-hp locomotive, built by the American Locomotive and General Electric companies will be placed in operation on a demonstration basis next spring by the Union Pacific Railroad. Present fuel is bunker "C" oil but it is hoped that coal may be used in future designs.

PHOTOELECTRIC control is built into new GE street lighting units. Designed for use on multiple circuits each luminaire is adjustable to operate between 1 and 6 foot-candles. Electronic unit is housed in a glass hood at top of the unit.

MACHINE DESIGN

Rocket Motors



...practical design problems and their solution

By George P. Sutton
Supervisor Propulsion Development
North American Aviation Inc
Los Angeles

RIGOROUS operating conditions of rocket motors pose engineering problems which make stringent demands on the designer. He has to allow for high heat transfer across stressed walls, establish the level of precision of fabrication tolerances, provide for high pressures, and keep the weight of the unit to a practical minimum. This article illustrates several of these design problems and indicates some solutions. Discussion is confined to liquid-propellant rocket motors, which can withstand repeated and prolonged firing.

Thrust of a rocket unit is derived from the momentum of ejected hot gases. Rocket propellants, i.e., a fuel and a liquid oxidizing chemical (called an oxidizer) are burned at high pressure in a combustion chamber and the resulting product gases are accelerated to a high velocity in a supersonic nozzle. Flame temperatures are on the order of 5000 F and the gas velocities obtained usually exceed one mile per second. The theoretical relations which determine the design conditions and the required size and shape of the rocket motor can be found in various texts, two of which are listed at the end of the article.

PRECISION REQUIREMENTS: *Fig.* 1 shows the basic

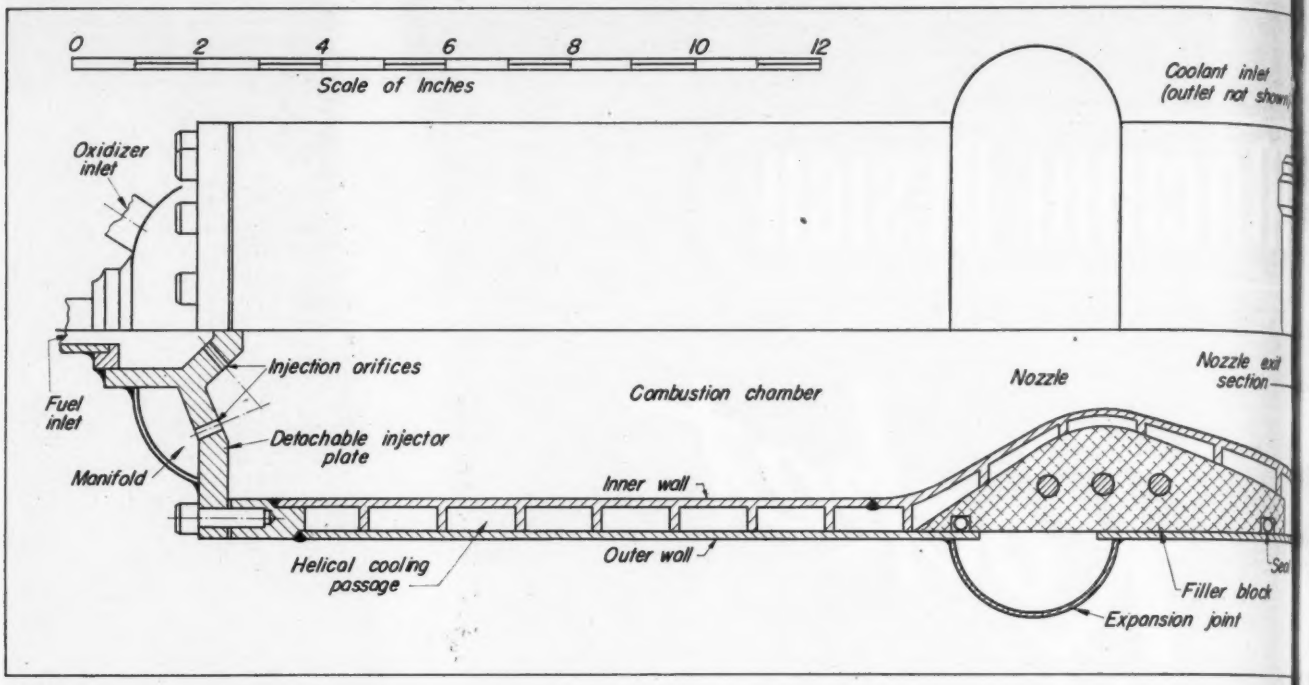


Fig. 1—Basic components of a 1500-pound thrust liquid-propellant rocket motor

components which make up a liquid propellant rocket motor. Even though it has no moving parts, a liquid propellant rocket motor is a piece of high precision equipment. The intense heat transfer and the requirements of accurate thrust and high efficiency impose severe fabrication tolerances on the rocket motor. The following have to be held to close tolerances, in part to insure interchangeability of individual rocket motors, and in part to prevent a malfunctioning of the unit; Nozzle throat area, the nozzle divergent cone alignment, the injection pattern, the cooling flow, and the surface roughness.

Nozzle Throat Size and Cone Alignment: Rocket motors for propelling projectiles, missiles and airplanes must have an accurately predictable thrust direction and magnitude. Thrust is approximately proportional to the throat area. As an indication of how the throat area controls the magnitude of the rocket thrust, a variation of 0.001-in. in throat diameter of a motor having 300-lb thrust capacity changes the thrust by approximately one pound. If the direction of thrust is not accurately aligned with the center of gravity of a freely flying and uncontrolled body, then its trajectory cannot be predetermined. For a projectile, for instance, the range and target hitting accuracy depend on this alignment. The fabrication requires aligning of the axis of a metal cone with the rocket motor assembly (frequently within one or two seconds of arc) and then aligning the rocket unit with the missile.

Injection Pattern: Combustion efficiency is known to be a function of injector design. Any manufacturing deviations from the injection design will cause appreciable differences in performance. One favorite injection pattern uses the impingement of a fuel stream against the oxidizer stream as a means of atomizing and mixing the propellants, *Fig.* 1. In this design the fuel and the oxidizing agent have to be injected through holes, the center lines of which intersect within a few thousandths of an inch. Fabrication of inclined holes with accurately intersecting centerlines requires special production and inspection equipment. It has been established that the performance drops some 10 per cent if the streams intersect only partially (with the centerlines of the holes less than the stream diameter apart) and as much as 50 per cent if the streams miss entirely. The injection holes usually act as orifices to control the propellant flows and therefore the ratio of the fuel to the oxidizer. It is not uncommon to hold these orifices to a total tolerance of 0.0005-in. in diameter.

Propellant Used for Cooling

Coolant Flow: One of the propellants, either the fuel or the oxidizer, is circulated through the cooling jacket before injection. Proper absorption of transferred heat by coolant, which is circulated through the regenerative cooling jacket, is necessary to keep the walls at a sufficiently low temperature to provide structural strength. Heat transfer rates are extremely high and critical in rocket motors, and if the coolant velocity at any one place in the jacket is lower than the safe design value a breakdown of the heat transfer and a melting of the wall will occur. Hence if the cooling jacket flow cross section is larger than the design value, the flow velocity will be lower and the chances for burn out will be increased. Fabrication tolerances on the cooling jacket have to be designed accordingly. This requires close tolerances on the internal diameter of the outer wall and on the spacers and the outside diameter of the inner wall. Allowance has to be made for the thermal expansion of the inner wall and the elastic deformations of both walls to determine the dimensions of the actual jacket passages when the motor is operating.

Surface Condition: The surface of the internal rocket wall has to be clean and uniform in roughness. In general, a sandblasted surface is just as good for heat transfer as one which is polished or finish machined. However, it is important that the surface be free of local protrusions or dips, such as welds, holes, scale, chips, etc. These discontinuities disturb the boundary layer and cause a locally high heat transfer rate resulting in failure of the wall.

To avoid these protrusions in welded assemblies, it is necessary not only to grind off the excess weld but also to match the mating pieces very accurately before welding to obviate a sudden step, *Fig.* 2. The most critical regions for these disturbances are those of high gas velocity, namely the divergent, throat and convergent sections of the nozzle, and the chamber walls immediately upstream of the nozzle. Near the injector the gas velocities are low and the disturbance is not as significant.

Heat Flow Intensity Creates Problems

EFFECTS OF HEAT TRANSFER: Heat transfer across the inner wall of rocket motors is extraordinarily high, because the burning gases are really hot (around 5000 F) and because the gas velocities (500 to 5000 ft per sec) are very much higher than those in conventional combustion equipment. The heat transfer rates in rocket motors measure from one to five Btu per sq in. per sec, which is 100 times larger than those of conventional combustion equipment. Heat transmission is most intense in the nozzle throat and the nozzle convergent region, as can be seen from the schematic diagram of *Fig.* 3. This is the cause for a number of effects on the rocket motor wall such as:

1. Thermal expansion of all components
2. Weakening of the wall material with high temperature
3. Thermal stresses in addition to the stresses imposed on the wall by chamber pressure and cooling jacket pressure

Thermal Expansion: The inner walls expand axially and radially since they are heated by hot gases. The German V-2 motor, for example, expanded approximately ¼-inch axially and 3/16-inch radially. Provision for axial expansion is usually made in the form of simple expansion joints, such as the bead in *Fig.* 1. Expansion joints to take radial deformations are generally not provided, since they are too complicated and too heavy to be reliable. Radial expansion, therefore, contributes to higher internal stresses. Because of high wall temperature (500 to 1800 F) the physical strength of the wall material is less than the strength at room temperature. The inside surface is the hottest and a temperature gradient is established across the wall.

Stresses in Rocket Motor Walls: The loads im-

Fig. 2—Examples of good and poor surface condition and welding of rocket motor inner wall

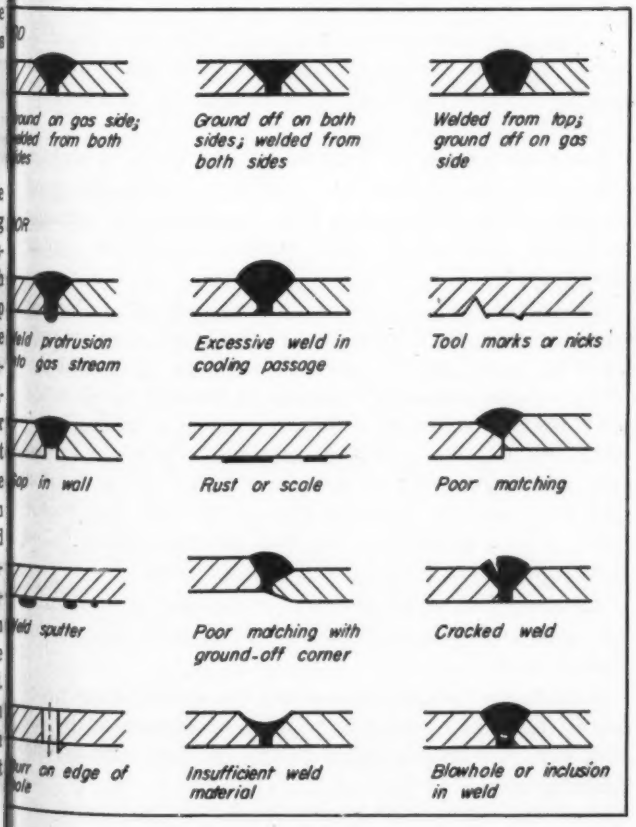
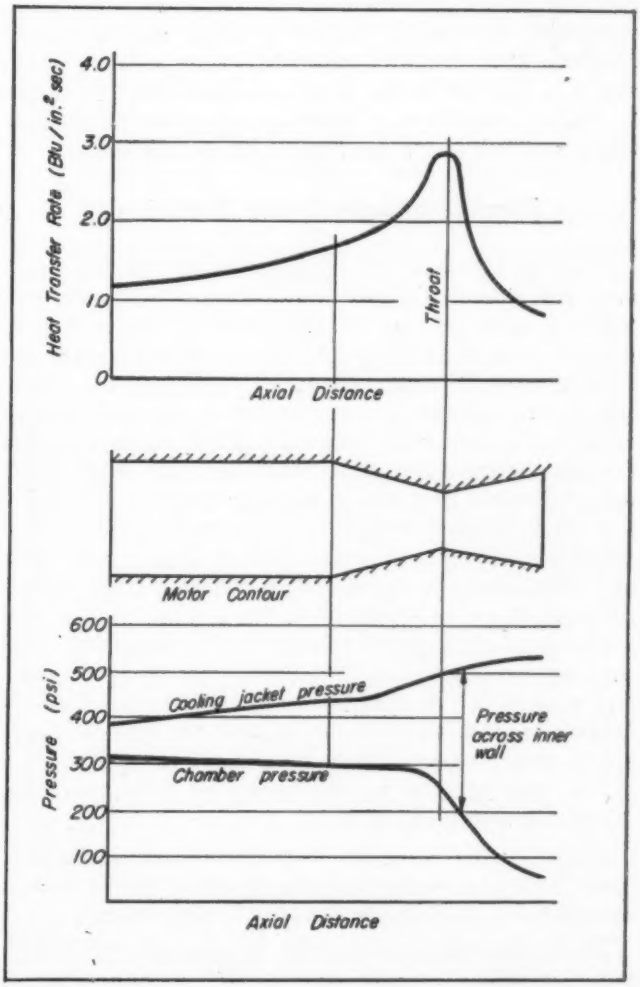

Ground off on both sides; welded from both sides
Welded from top; ground off on gas side
Excessive weld in cooling passage
Tool marks or nicks
Rust or scale
Poor matching
Poor matching with ground-off corner
Cracked weld
Insufficient weld material
Blowhole or inclusion in weld

Fig. 3—Typical heat transfer and pressure distribution along the length of a rocket motor



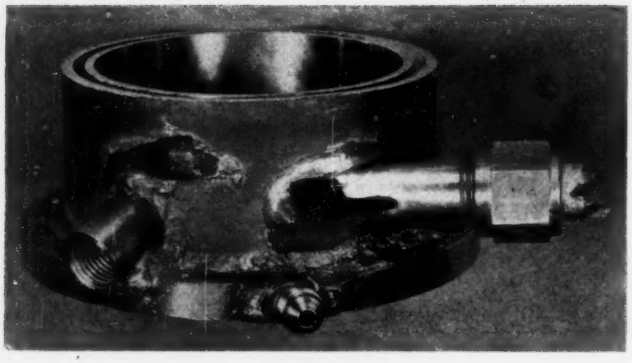


Fig. 4—Burned out portion of cooled chamber on rocket motor. Failure was caused by poor weld between inner wall and boss. Hot gases burned through both walls

posed on the inner wall of the rocket are:

1. Gas pressure in the chamber or the nozzle
2. External pressure in the cooling jacket
3. Axial loads, (e. g., thrust or pressures acting on surfaces normal to the motor axis
4. Loads imposed by accelerations of the whole missile or aircraft
5. Internal loads imposed by thermal deformation
6. Transient pressure surges during starting or stopping.

Difference between the internal and external pressures (items 1 and 2) gives a compressive stress in the wall and therefore a considerable buckling and collapsing tendency. Motor failures attributed to buckling of the wall are common. This buckling tendency is highest in the divergent nozzle section, *Fig.* 3. Items 3 and 4 give stresses which are relatively small and can therefore frequently be neglected.

Thermal Stresses Cause Distortion

Thermal stresses will be proportional to the temperature differential across the wall, and depend on the properties of the wall material. These stresses are a maximum at the hotter, inner surface (compression) and at the relatively cooler, outer surface (tension) and a minimum in the center of the wall. Their magnitudes frequently exceed the yield, causing a permanent deformation. The inside surface, being hottest, will yield first and cause a permanent deformation and residual stresses in the wall. This compression yielding is the reason for the peculiar shrinkage phenomenon encountered in rocket motors; their internal diameter (notably in the throat region) diminishes slightly with each successive operation and the cooling passage is frequently distorted. Transient surges in the chamber or cooling jacket depend largely on the injection design and the starting and stopping procedure. They require the use of appropriate load factors on the pressures acting on the inner wall shown in *Fig.* 3.

SELECTION OF WALL MATERIAL: A material for the inner wall of a cooled, liquid propellant rocket motor should have the following desirable properties:

High thermal conductivity to permit the ready transfer of heat to the cooling fluid and to allow a more nearly uniform wall temperature with low thermal stresses and a minimum of distortion. It reduces the tendency to produce local hot spots.

Fig. 5—Inner wall of nozzle with machined helix for controlling flow of the coolant

High yield and ultimate strength at elevated temperatures to make a light-weight construction feasible. Rocket motors are airborne equipment and weight is therefore at a premium.

High melting temperature to prevent failure of the material if local overheating does occur.

Simple fabrication method, an important requirement, often overlooked. This includes such properties as ease of forming, weldability, and machinability of the material.

Good resistance to heat shock, to allow a sudden application of heat during starting. Although most metals are satisfactory in this respect, many ceramic materials will crack when used as a rocket motor liner.

Resistance to erosion and to chemical attack by hot combustion gases. Most rocket combustion gases produce an oxidizing atmosphere and the chemical activity at elevated temperatures is greatly accelerated over that at room temperature. Once a metal is locally melted, the combined effect of chemical oxidation and high velocity gases acts like a blow torch and very quickly burns a large hole into the motor wall. *Fig.* 4 shows a motor and injector which failed in this manner, because of a faulty weld.

Low coefficient of thermal expansion to minimize the design problems associated with the expansion of heated parts.

Resistance to vibration, since the combustion process in a rocket motor is usually accompanied by vibrations covering a wide range of frequencies and amplitudes.

Unfortunately, no known material possesses all

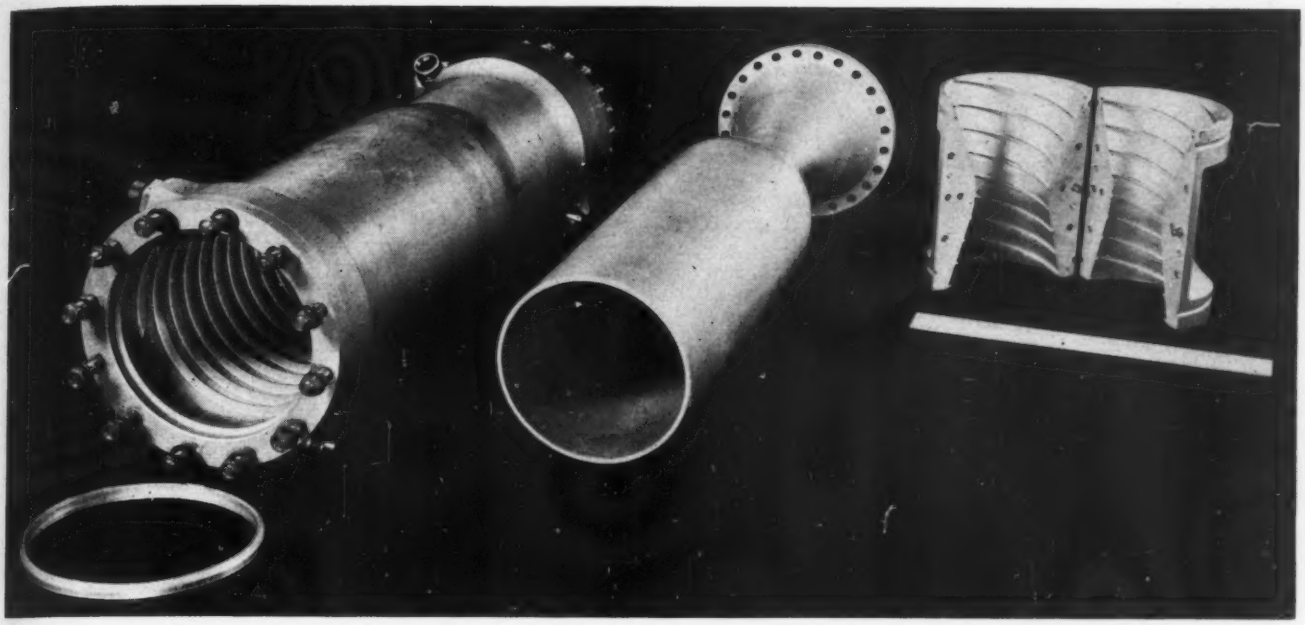


Fig. 6—Typical machined rocket motor disassembled. No welds are used, all parts being bolted together

these characteristics. High thermal conductivity, high strength and high melting temperature usually are not encountered in the same material. For example, aluminum and copper have high thermal conductivity, but lack high strength at elevated temperature and melt readily. On the other hand, stainless steel and a number of special turbine blade alloys have excellent strength at 1000 to 1800 F, but their conductivity is usually less than one third that of aluminum, giving rise to high surface temperatures.

Although copper, aluminum, stainless, alloy and low carbon steels have been used with success, each present some limitations in application. It is, for example, impossible to construct a very large motor of stainless steel with conventional designs, because the required heavy wall thickness would not only induce excessive thermal stresses in the walls, but would make local melting highly probable. Copper motors, for instance, are almost always extremely heavy.

A desirable wall material would be a combination consisting of a coating of high temperature material on the gas side, on top of a base of a steel or copper alloy. Research on the application of ceramic coatings or on sandwich construction might give such a favorable combination.

FABRICATION METHOD: One common fabrication method is to machine chamber, nozzle and injector parts from bar stock, blocks, pipes, or plates. This method gives accurate parts and is preferred for the fabrication of small motors up to 6 or 10 inches in diameter, but is not too practical for large motors. The machine set-ups can become relatively complicated as exemplified by the spiral guide spacer, which was machined onto the diverging-converging nozzle shown in *Fig.* 5. The individual pieces are then joined by welding or flanges. The rocket motor shown in *Fig.* 1 is fabricated in this manner. For medium sized and large motors rough forgings and, occasionally, castings are used; these are then machined to finish dimensions, *Fig.* 6.

Use of large castings has not been favored generally for rocket motor construction, because the required section thicknesses are usually thinner than those dictated by good casting practice and because the properties of cast materials are frequently inferior to those of similar wrought materials. The use of a cast injector, in one instance, became impossible because of porosity between the fuel and the oxidizer passages.

Formed and Machined Parts Combined

Formed sheet metal and plate offer a satisfactory method of fabricating large motors which is light and simple. The accurate shaping of nozzle and chamber components with double curvature presents many headaches, particularly the forming of heavy gage sheets, e.g., ¼-inch thick, to a predetermined cooling passage curve within a small deviation from the intended line. A combination of formed and machined parts is in common usage.

Welding requirements of rocket motors are severe. Only good welds, which are pressure tight, strong, have a smooth contour, and can take heat shock and deformation can be tolerated. Inclusions or cavities in the weld obstruct the flow of heat and cause hot spots; weld cracks usually open up under vibration and thermal distortion. Importance of these high quality welds might be illustrated by the fact that during World War II the Germans would select a good welder, give him a special instruction course, subject him to examinations and make him produce satisfactory samples before letting him weld on the V-2 rocket motors. In addition, all welds were pressure-tested and X-rayed.

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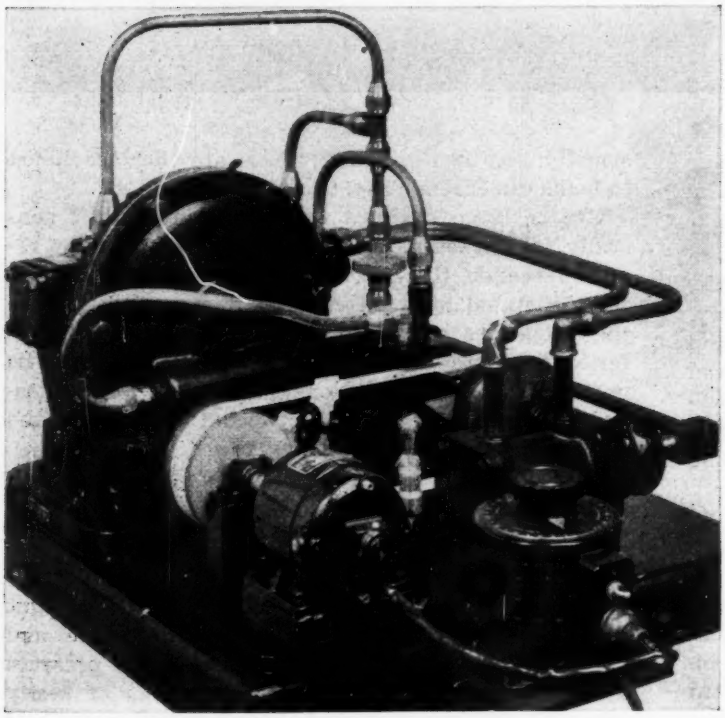
M. J. Zucrow—*Principles of Jet Propulsion*, John Wiley & Sons, Inc., New York, 1948.

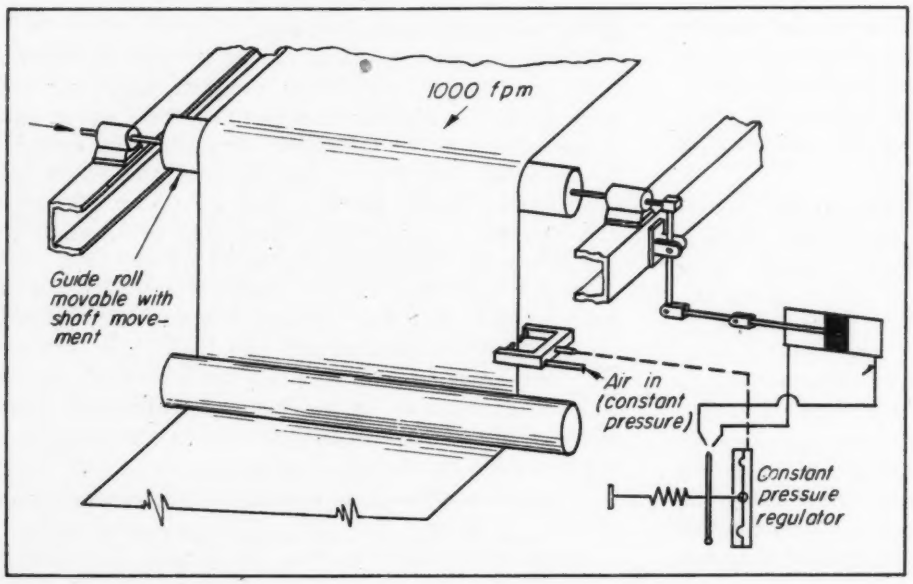
Scanning THE FIELD for Ideas



EDGE position of sheet being processed over rolls can be controlled within limits as narrow as plus or minus 0.05-inch with the air-jet control illustrated at left. Any process in which coating, coiling or an endless belt conveyor is used may be controlled by this device designed by Askania Regulator Co. Two air nozzles are used to pass a narrow stream of air past one edge of the sheet as indicated in the schematic drawing left-below.

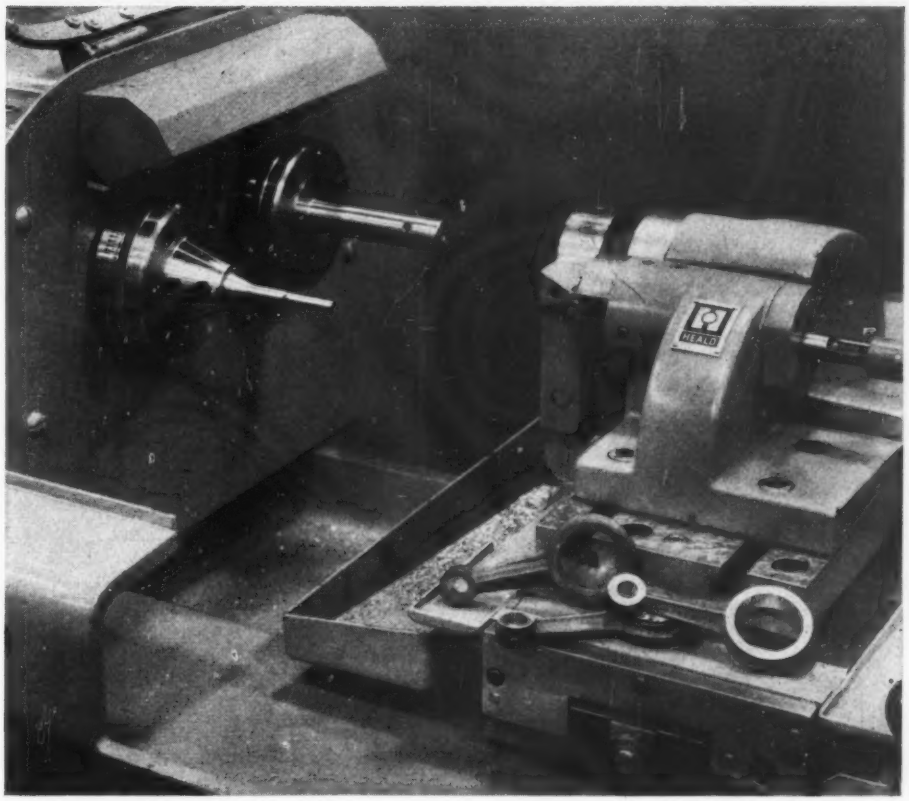
Air from one nozzle is received by the other, the receiving nozzle being connected to the diaphragm of a standard pressure regulator. Moving between the nozzles, the sheet cuts a certain amount of the air flow to give an impulse to the regulator. This regulator supplies power to a cylinder which changes the position of the sheet on the roller either by pivoting the axis of the roller, or by moving it from side to side when crowned rolls are used. When the sheet moves in response to this movement, it cuts off more or less of the air flow, changing the impulse on the regulator until a control position is reached.

Position of the air nozzles can be changed with a handwheel in order to change the desired position of the sheet in relation to the center of the roll. When the air nozzles are moved to one side or the other, the sheet will follow until the proper control impulse is received.

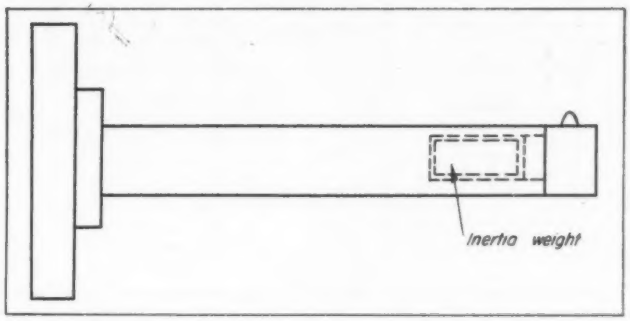


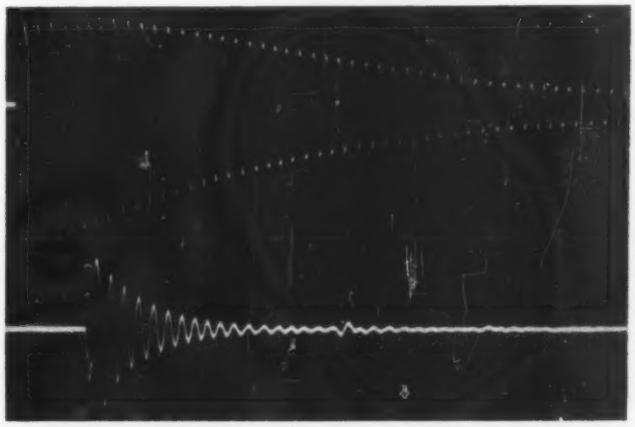
Reducing vibration in machine tools for precision finishing, particularly boring

machines such as illustrated at right, has been a headache because vibration causes chatter marks on the finish. The phenomenon is a self-excited vibration in contrast to the more common forced vibration usually found in machinery and is particularly troublesome with boring bars having a high ratio of length to diameter.



Frequently such chatter can be eliminated by increasing the damping, i.e., the rate at which vibratory energy is dissipated from a boring bar. An effective damper consisting of a cylindrical weight fitted loosely into the end of a bar is shown, right-center, diagrammatically. In this damper it is essential that the clearance space around the weight contains the proper damping fluid, usually air, and that the diametral clearance and mass of weight be suitable for the dimensions of the quill. With the proper proportions, vibrations die out rapidly as evidenced in the "before and after" damping oscillographic records shown, right-below, resulting from research and development at the Heald Machine Co.

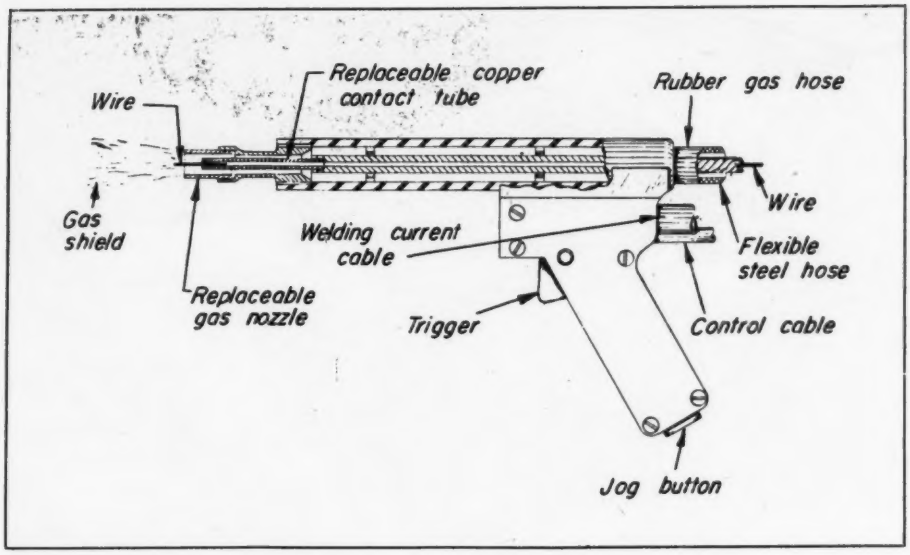


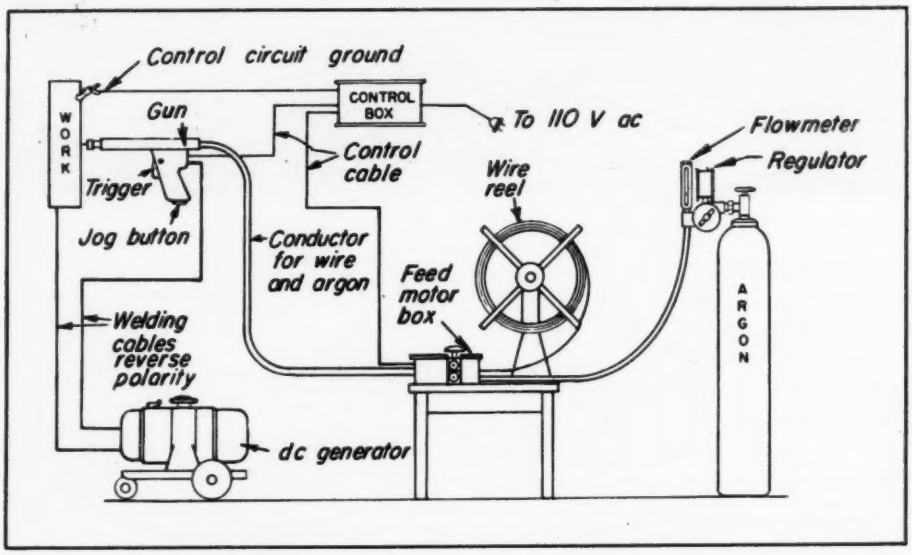


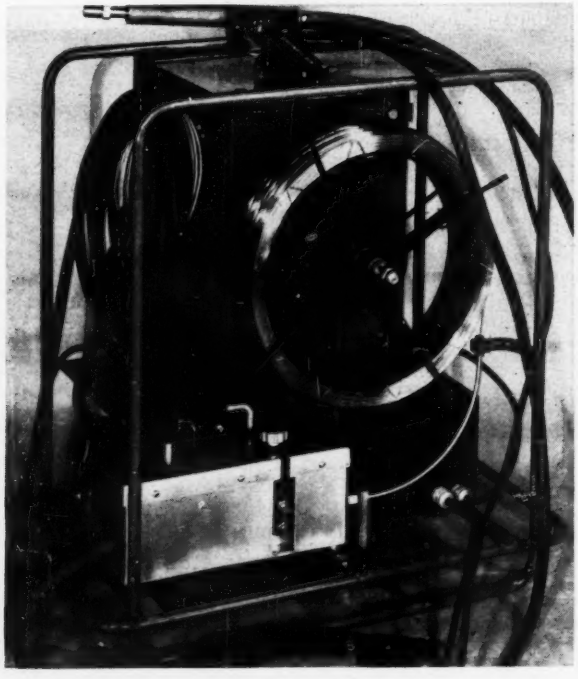
Quills have been made to bore satisfactorily with length to diameter ratios of 8:1. Damping effectiveness, however, decreases with extreme bar length due to lower frequency vibrations and consequent inability of the weight and damping fluid to absorb or dissipate the slow vibrations.

Operation of this damper is simple. Quill vibration will displace air from one side of the clearance space to the other around the weight which tends to remain at rest because of inertia. This displaced air in the course of its flow from one side to the other dissipates energy by virtue of its viscosity, causing vibrations to die out rapidly. If the coupling between the inertia weight and the quill is too strong, the weight will move with the quill and fluid will not be displaced from one side to the other. If coupling is weak, fluid flows too freely.

Gas-shielded, metal-arc, manual method of welding has been developed for joining heavy sections of aluminum and aluminum alloys. In this method, developed by Air Reduction Laboratories, the metals being welded are shielded from the air by argon gas. Engineering problems and the development of equipment have involved new concepts of current density, gas shielding and filler-wire feed speeds. Mechanical properties of alumi-







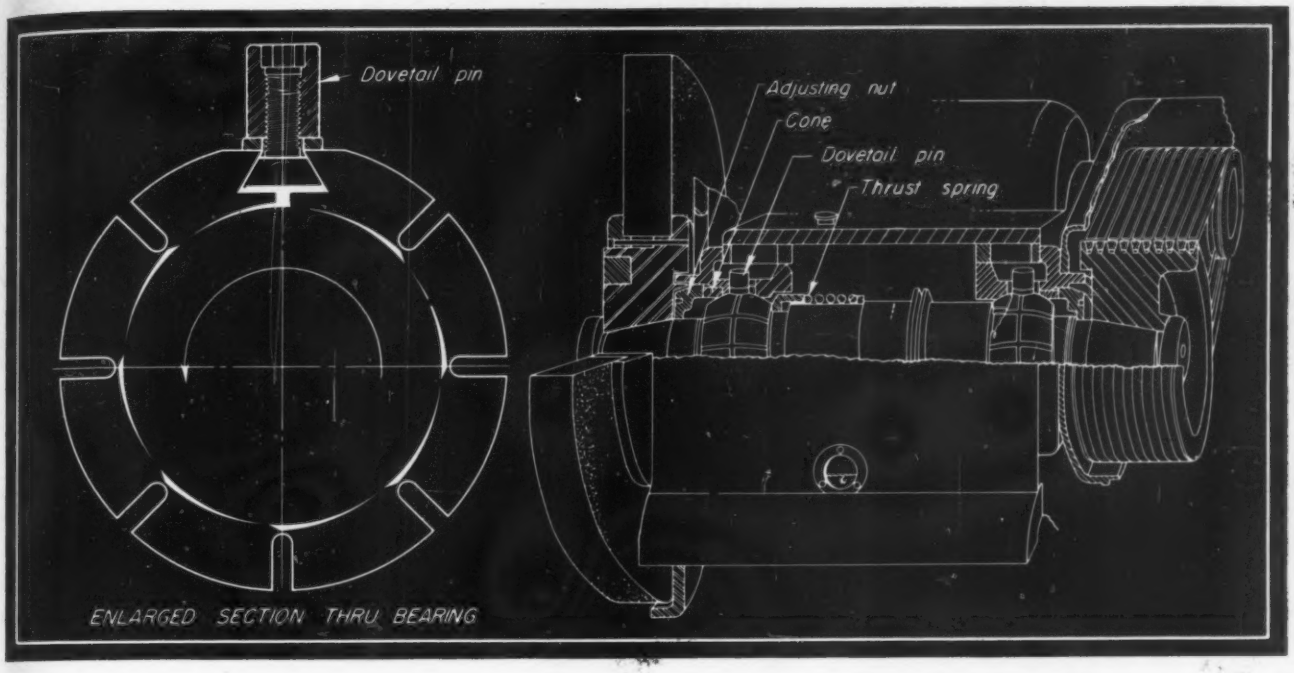
num welds are equal to the properties obtained in castings of the same analysis as the filler metal.

Consumable wire, replacing the conventional nonconsumable tungsten electrode, is fed through the barrel of the welding gun shown in sectioned view at left. Power is supplied from a standard d-c welding generator shown in the schematic diagram, left-center, of the welding method. Component parts for the equipment, left-below, are the welding gun, argon hose, concentric with which is the metallic tube for conveying the welding wire, a three-conductor control cable, feed-motor box, control box and wire reel.

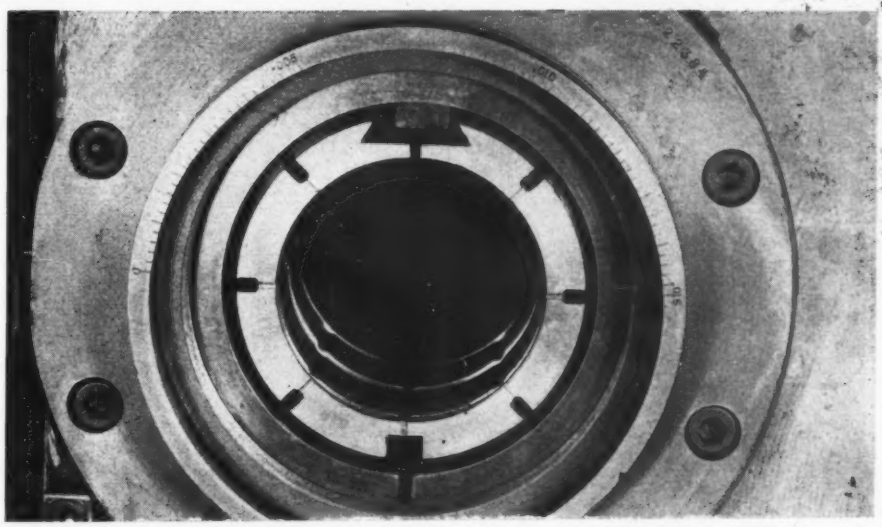
Gas hose and wire conductor are connected to the feed-motor box; the control cable also connects to this box which contains a governor-controlled motor, speed reducer, feed rolls, and a solenoid valve. The control box, which is connected by the three-conductor control cable to both the gun and the feed-motor box, contains the relays which control operation of the wire feed. It also houses the transformer for the control circuit.

Trigger on the gun readies it for welding and the jog button feeds wire as may be required when not welding. When arc voltage is established the controls operate to energize the feed motor and deliver filler wire. Current densities twelve times those used in ordinary arc welding have been successfully employed. With such high current densities it become necessary to provide high rates of filler metal feed in order to make satisfactory welds. Filler wire speeds range from 100 to 800 inches per minute. The arc obtained may be described as a nonsensitive arc, in that changes in arc length result in smaller changes in arc voltage than encountered in normal welding.

Adjustable sleeve bearing is self aligning and also serves as a thrust bearing on Landis Tool Co. precision grinders. Top of next page shows a bearing in its cartridge but with the shaft and adjusting nut re-

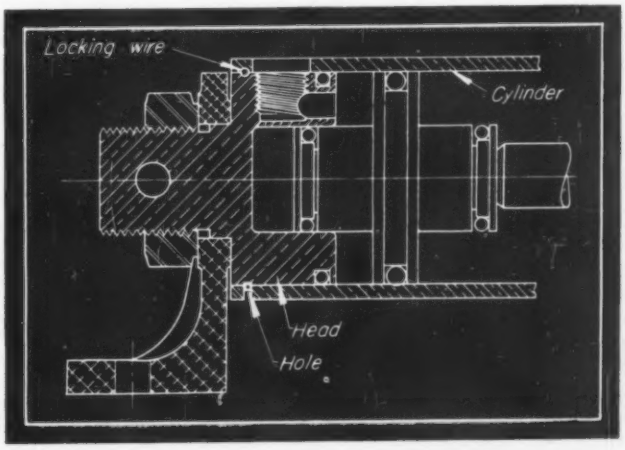


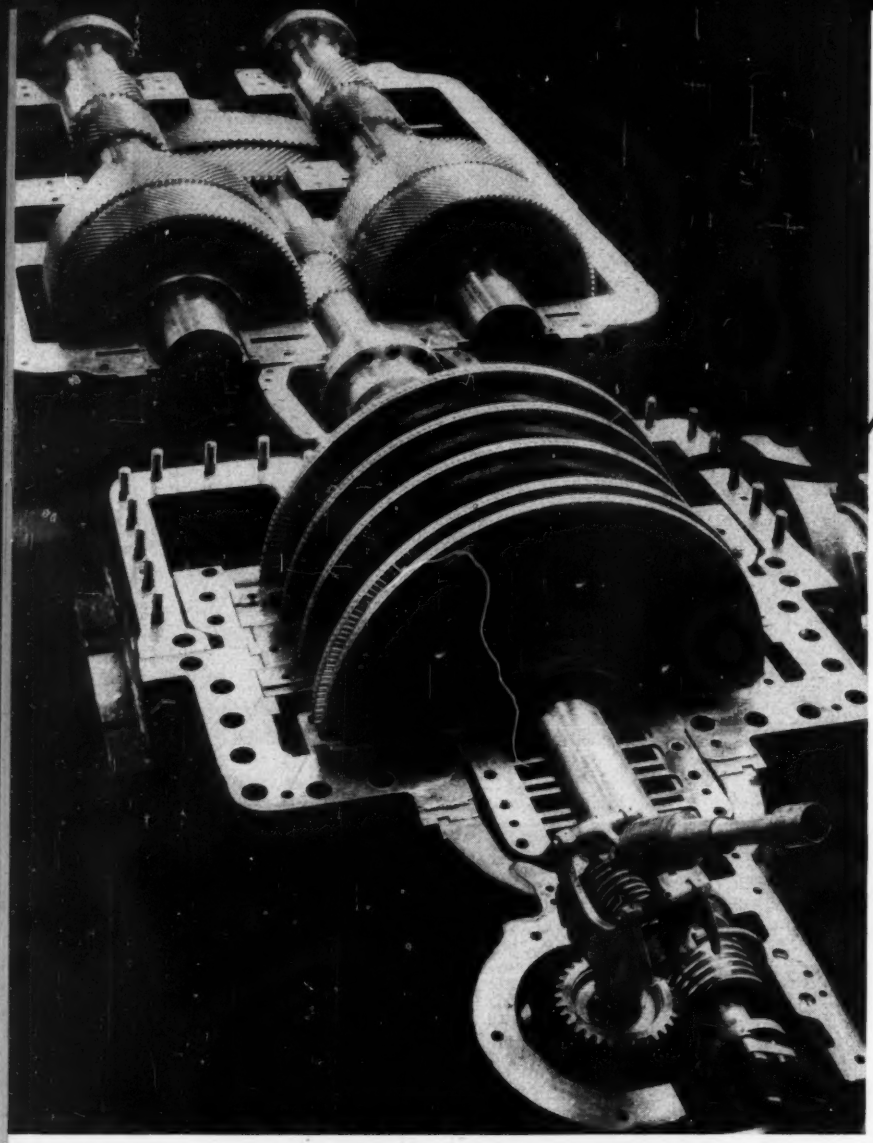
moved to show the features of the bearing. Graduations on the end of the housing are reference points for use when setting up or changing clearance of the bearing. The bearings are generally set with close clearance between the bearing and spindle. On some of the smaller Landis machines, the bearings have a clearance of less than 0.001-inch when running warm.



Bearings are spherical type and short in length to assure alignment regardless of spindle deflection or housing distortion. A dovetail groove at the split section holds an adjustable dovetail pin which both retains the bearing against rotation in the housing and serves as a clearance stop, opposing the closing action of the cones as shown in the sectional view at right. Deep radial slots in the bearing assure that, on takeup, it will remain concentric. Bearings are steel backed, centrifugally poured, babbitt lined with babbitt lined faces for thrust bearings.

Simple locking design for assembling a head to its cylinder is shown in the drawing at right for a hydraulic cylinder built by Modernair Corp. The head is held in place by a locking wire which is fed into place through a hole in the cylinder by turning the head one revolution. A groove is cut, half the thickness of the wire, into both the head and the cylinder so that when the wire is in position, the head is held securely.





ked-Train G

. . . offers increased torque capacity, reduced overall size, and variety of drive component arrangements

By R. L. Benford
Gear Engineering Division
General Electric Co.
Lynn, Mass.

Fig. 1—Typical locked-train, double-reduction gear coupled to steam turbine is shown here on test stand

MANY advantages can be obtained in some types of gear sets by using multiple engagements instead of the usual arrangement of a pinion and one mating gear. Among the chief benefits of multiple-engaged helical gears, *Fig.* 1, are reserve capacity, in-line drive, effective use of space and weight, and variety in machinery arrangement. Most multiple-engagement gears fit the definition for locked-train gears and consideration of several factors is important for the successful exploitation of such gear sets. While helical gearing is generally used, most advantages of locked-train gearing apply to all types of gears, such as spur, worm, bevel, etc.

Locked trains considered here are defined as arrangements of shafts and gears, with an input shaft connected by multiple sets of intermediate shafts and gears to the output shaft, the transmitted power being divided among the intermediate sets by multiple-tooth engagements on the input gear and the output gear. Power is distributed from a single source through several branches, and is recombined into a single output.

Since the stresses in a gear tooth occur only during engagement, the torque capacity of a gear wheel or pinion may be multiplied by using multiple engagements. Referring to *Fig.* 2, it is evident that by adding another gear wheel, the output torque of the pinion can be doubled without any increase in tooth loading, because the pinion can make two distinct engagements per revolution.

Multiple engagement simultaneously puts to work more of the teeth on a pinion or gear, and reduces the interval when the teeth are "just riding." The idea has its counterpart in many practical and familiar arrangements. Two horses can pull twice as hard as one; a four-engined bomber or a double-headed train can go faster and pull harder than a single engine will allow; a multiple-disk clutch or multiple-runner ship's thrust bearing can multiply the capacity of machine components in a compact manner; and a splined shaft with many teeth has several advantages over a shaft with a single key.

In these examples, when one horse or one engine quits, when one clutch disk or one bearing surface

Gearing

fails, there remains a reserve for limping home under reduced power. With gears designed to run at their short-life capacity, as they sometimes are, this reduces the hazard of total failure.

The commonest arrangement of locked-train gears is shown in *Fig.* 3. All gears are in one plane and the input and output shafts rotate in the same direction and in line. The two intermediate shafts, each with its high-speed gear and low-speed pinion, rotate in the opposite direction, each transmitting half the power from the input shaft to the output shaft. This is the basic design of the "twinned" double-reduction gear for relatively high ratios of speeds.

Planetary gears are almost always built with multiple planets to give a symmetrical planet carrier and reduce the overall size by increasing the capacity of the sun pinion and ring gear. *Fig.* 4 shows a common arrangement using three planets. Airplane engine builders often use as many as twenty planets between the sun pinion and the ring gear, thus reducing the size and weight of the reduction gear between the engine and the propeller.

Another example of twinned gears, this one with two idler gears connecting a high-speed pinion and a bull gear is shown in *Fig.* 5. With this arrangement, the pinion and bull gear could be reduced in size for a given capacity and the two idlers fitted into convenient spaces, resulting in smaller overall size and weight.

Ship propulsion gears, which have the highest power ratings, must drive ships for hundreds of thousands of miles and withstand severe short-period overloads. During World War I, hundreds of merchant ships were fitted with reduction gears similar to those shown in *Fig.* 6. This arrangement differs principally from that shown in *Fig.* 3 in that the large bull gear is not in line with the high-speed pinion and turbine. Other refinements, some of which are shown in *Fig.* 6, will be mentioned later. These ships, with their single-turbine locked-train gear drives, performed very capably.

Most large American naval vessels are equipped with multiple-turbine double-reduction gears of locked-train design. The bearing seats for the intermediate shafts and high-speed pinions are rigidly supported by the sloping members of an A-frame structure. With pairs of low-speed pinions meshing on each upper side of the bull gear, a very sensible arrangement of high-pressure turbine, low-pressure turbine, and condenser results. In fact, this gear design has allowed the greatest engine efficiency with the least weight and space. Its performance during World War II was extremely gratifying. The driving pinion with its center on the line between gear centers, shown in *Fig.* 7, has virtually no bearing reaction because of the balanced forces. With equal load distribution to each gear, the bearing supports only the pinion weight, a minute percentage of the driving forces involved. Thus, smaller journals can be used.

Some advantages of twinned or locked-train gears are common to all arrangements, although a few apply only to certain designs, as mentioned previously.

1. Multiplied torque capacity for a given pinion
2. Reserve capacity in event of partial breakdown
3. Concentric and symmetric drive
4. Rigid casing support
5. Effective utilization of space and weight
6. Floating high-speed pinion and low-speed gear

Fig. 2—Below—By driving two gears from a pinion the torque capacity of the pinion can be doubled without increasing tooth loading

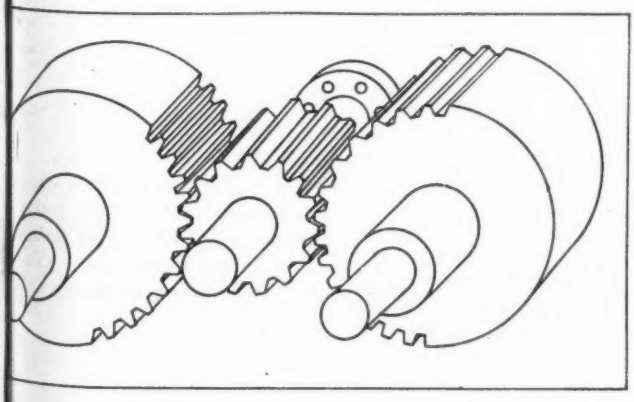
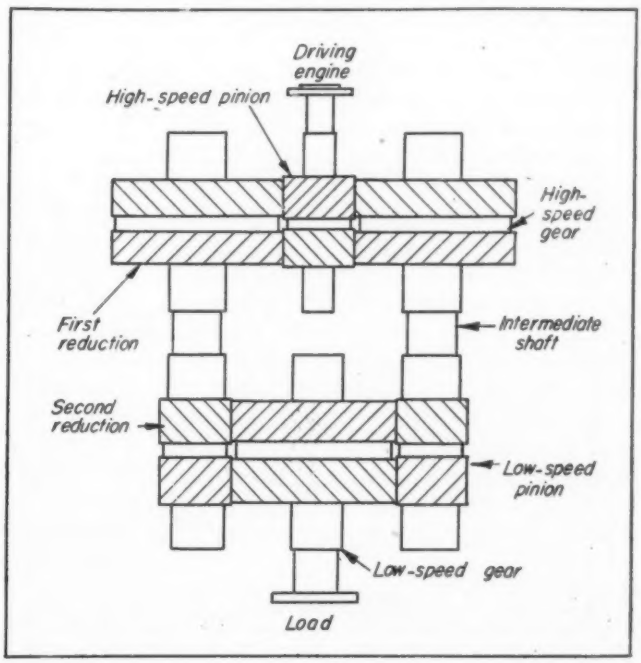


Fig. 3—Basic design of "twinned" double-reduction gear for relatively high ratios of speed. In this design all gear axes are in one plane



7. Advantageous machinery arrangement.

Like their counterparts in other devices, locked-train gear systems require equal load distribution among their multiple branches. A team of horses is harnessed to its wagon by a double tree; four-engined bombers or double-headed locomotives require constant adjustment and synchronization if they are to share the load efficiently; multiple-disk clutches, mutiple-runner thrust bearings, and splined shafts were used only after proper machining and assembly techniques were devised to distribute their loads evenly. Equal load distribution can be attained in locked-trains of gears by three basic actions:

1. Flexible intermediate members
2. Adjustment of position at assembly
3. Continuous automatic adjustment of position in operation.

Adjustments are necessary, of course, for all locked-train devices, and the positions of gear and pinion teeth, keyways, coupling bolt holes, and other pertinent angular locations cannot be machined without relative angular errors unless unusual tactics are used. In the class of gears considered, namely gears for the transmission of power, the machining tolerances would be insufferable and a slight error would make a part unfit for use if no means for nullifying the errors were provided. By providing some adjustments it is possible to disregard entirely all angular relationships during machining and yet assemble the parts so as to attain acceptable load distribution. This feature greatly reduces the cost of manufacture.

The potency of flexible intermediate members is shown by the curves of *Fig.* 8. For greatest effectiveness there should be a large displacement under rated torque. This displacement is usually torsional, as shown in *Fig.* 9, where the high-speed gear and low-speed pinion of each intermediate shaft have a large relative twist under load.

With an intermediate shaft of flexible design, as shown in *Fig.* 9, when one branch carries extra torque it twists more than the lagging branch. The second branch will then be repositioned at the mesh where the lag existed, and will begin to transmit some torque. If there is a lag at no-load, the two branches will never carry exactly the same load, since one shaft must twist more than the other to engage both. The curves of *Fig.* 8 give a quantitative value to these discrepancies.

It is apparent that by making the full-load twist

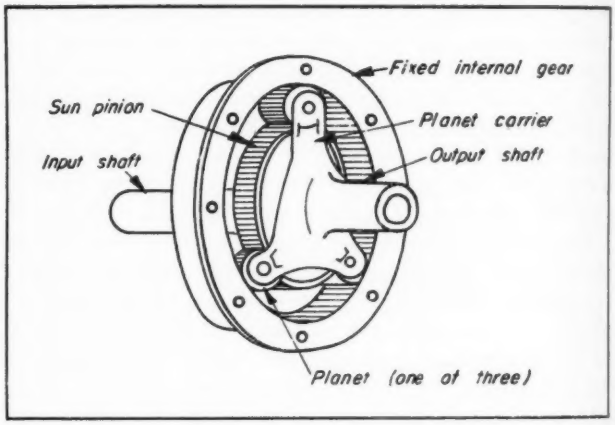


Fig. 4—Above—By using multiple planets in planetary gear trains the torque capacity of the sun and ring gears is increased

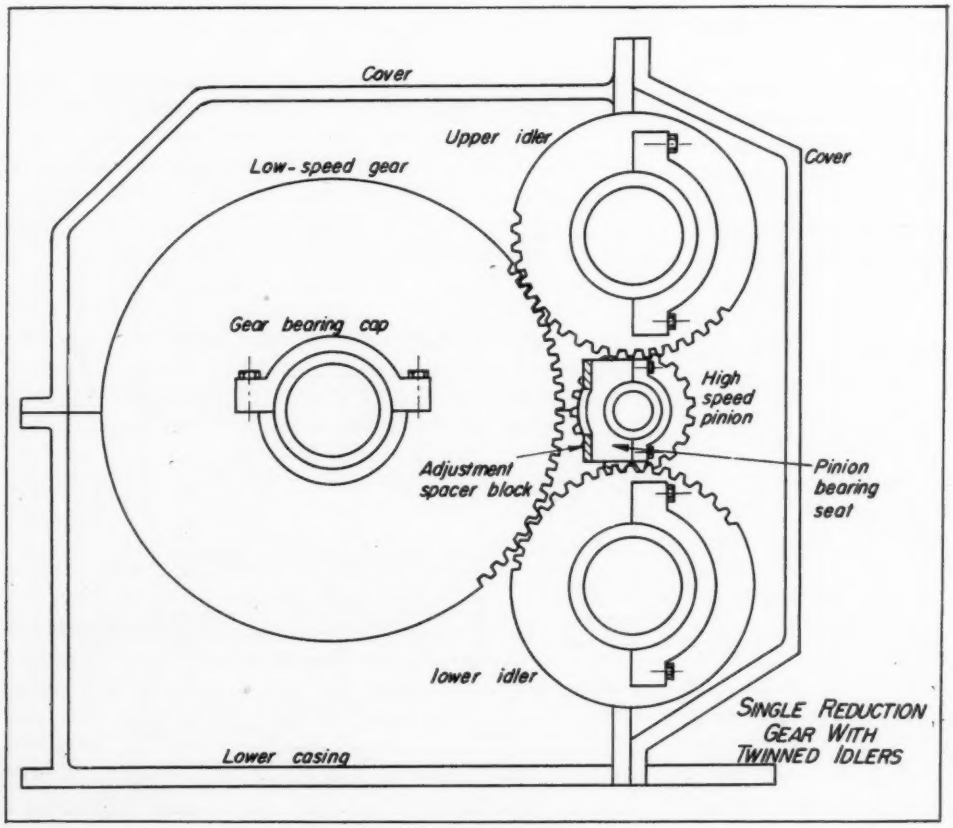


Fig. 5—Left—Use of two idlers between high-speed pinion and bull gear permits reduction in gear and tooth sizes for given capacity

Fig. 6 — Right — In this locked-train, double-reduction gear the turbine and bull gear axes are offset

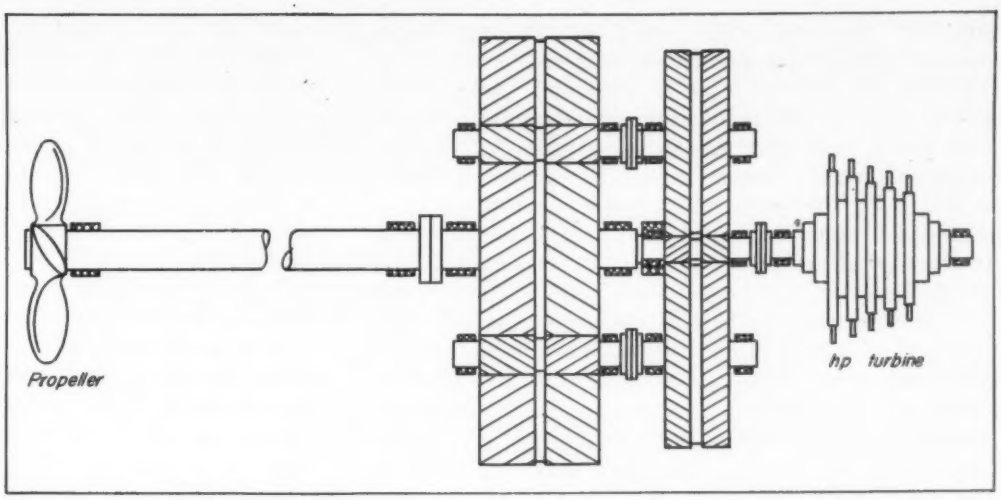


Fig. 7—Below—With the driving-pinion center on the line between gear centers, there is virtually no loading on the pinion bearing

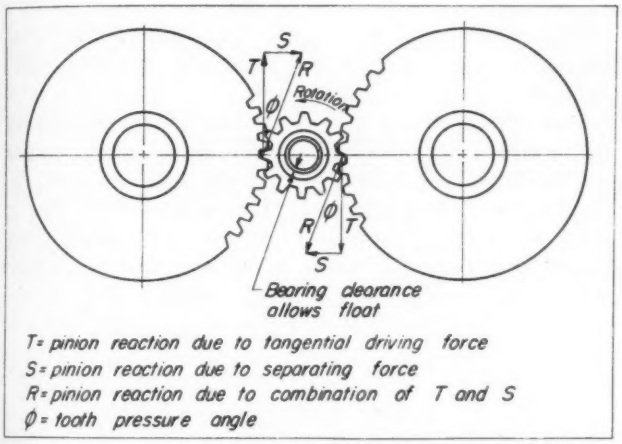
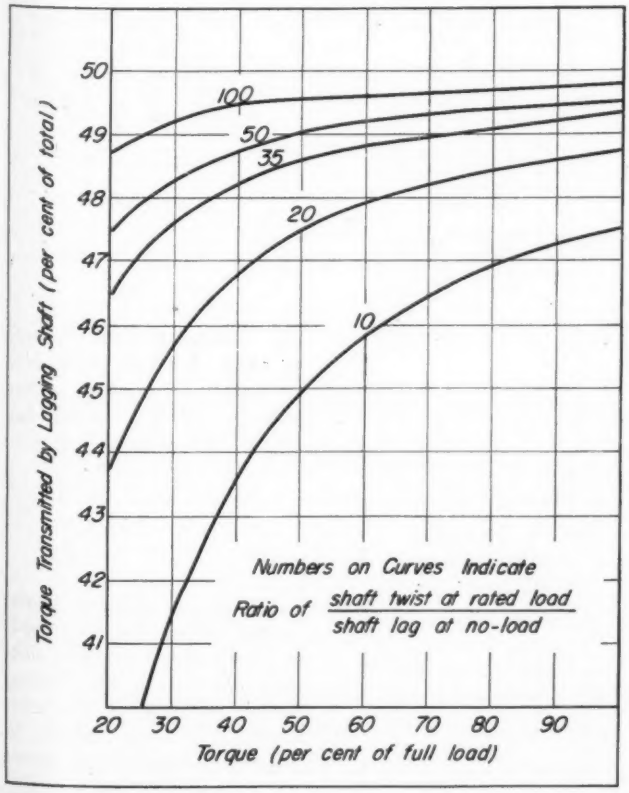


Fig. 8—Below—Torque distribution in locked trains of gears which employ two intermediates. Curves apply to lagging shaft and gears



of each side many times greater than the no-load lag on one side, the load split is practically equal at loads near the rating of the gear set. For example, if the intermediate shafts would advance their low-speed pinion teeth 0.050-inch by twisting due to equally distributed full-load torque, and were assembled with one shaft pinion lagging in mesh by 0.001-inch, the load distribution would be better than 51/49 at all loads above 40 per cent of rating. This is certainly acceptable, and is based upon a typical ratio of twist/lag.

To further insure a high value of twist/lag, means are always available to reduce the lag. Upon first assembling all the gear components in their bearings, the driving faces are pressed together lightly, except that usually, in a random assembly, all teeth will not engage properly. In a double-reduction gear of the form shown in *Fig.* 1, two classical methods are used to eliminate the lag at a tooth mesh.

In the first and oldest method, one intermediate coupling is disassembled and its hub loosened so it can be turned freely. Simulated torque is applied by wrenching various components until all ahead faces of the gears, pinions, and couplings are engaged. The loosened coupling hub has been turned somewhat on its shaft during this procedure and a careful measurement is made of the amount of turn. This usually is the offset between shaft and hub keyways, and is corrected by machining a special offset key.

A variation of this method is to postpone the machining of one hub keyway until assembly, where the correct position is scribed when the parts are being torqued. The hub is then slotted and a straight key used. Three or four men can do this job accurately and quickly on even the largest gear sets.

The second method, now in favor, uses the vernier action of pinion, gear, and coupling on one shaft. Lag can be measured initially either at the coupling keyway as before or at one low-speed pinion and gear mesh. It is usually better to measure lag at keyways, since a large lag would otherwise take up all the backlash and then wind up a shaft, giving a false reading.

After the lag has been measured, a table of corrections is referred to and the vernier action begun. The procedure is to rotate the pinion and gear and

coupling hub separately, each by an appreciable amount, but so that the net effect is very small. This can best be explained by an example: Assume the intermediate shaft is permanently assembled to its high-speed gear and that a keyed coupling hub is mounted on the opposite end. The low-speed pinion is driven by the shaft through the coupling.

Assume that the high-speed gear has 125 teeth, the low-speed pinion has 33, and the coupling 23. The shaft diameter at the coupling key is 5.25 inches and its circumference is 5.25π. Therefore, if the high-speed gear is turned one tooth while disengaged, its shaft will turn $(5.25\pi/125)$ inch, or 0.1319468-inch. Similarly, adjusting the low-speed pinion and intermediate coupling by one tooth will give relative displacements of 0.499798-inch and 0.7171026-inch at the keyways.

A little figuring shows that very fine corrections can be obtained by taking differences between multiples of these coarse corrections. Taking two elements at a time for preliminary attempts, rotating the high-speed gear eleven teeth moves the outer keyway 11 × 0.1319468-inch or 1.4514-inch, while rotating the coupling independently by two teeth returns the outer keyway 2 × 0.7171026-inch, or 1.4342-inch. The net movement is the difference, or 0.0172-inch.

As usual, in calculations involving small differences between large numbers, the numbers should have several extra figures right up to the final step. Pairs of movements, gear-pinion, gear-coupling, and coupling-pinion, are tried for small corrections, and their differences compared with each other for the refined adjustments. A small correction is obtained here by rotating the gear by one tooth, the pinion by six teeth, and the coupling by four teeth, giving a net movement of 0.0015-inch.

The correction tables are made beforehand in the engineering office, with enough corrections listed to guarantee acceptable value of twist/lag for the gear set no matter what the initial lag. Methods have been devised to perform the arithmetic in a direct and quick manner.

It should be noted that the second method involves no special parts or further machining, making it cheaper and more adaptable to assembling replacement parts in a ship, factory, or power plant. As before, the job can be done competently by a small crew working without bulky equipment.

The third action, continuous repositioning in operation, can be simple or involved. A simple correction, always available, is to allow the driving pinion room to float by increasing its bearing clearance. Then, as seen in *Fig.* 7, when one mating gear transmits appreciably more load than another, its reaction forces the pinion into mesh with the other gear within the limits allowed by the bearing clearance.

A more effective and more involved system is possible and practical when either the high-speed gear or low-speed pinion, or both, are single-helical. A thrust results when the shaft transmits load and there is a tendency for the shaft to move so as to drop its load. If the thrust bearing which restrains this motion is mounted in such a way that it can move slightly, and is connected by a linkage to a similar thrust bearing on the twin shaft, the thrust on each shaft should be balanced directly, thus balancing the load division.

The linkage can be a simple lever, or cables, or hydraulic pistons, or, for gears with more than two intermediates, gimbals or swivels. Such devices appear to be promising for use on small highly-loaded precision gearing where adjustments at assembly are handicapped by the small sizes and inaccessability. A general arrangement is shown in *Fig.* 10.

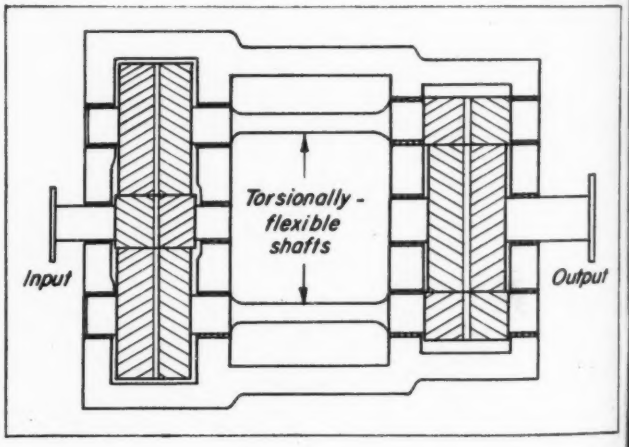


Fig. 9—Above—One requisite of equal load distribution in locked-train gearing is torsionally flexible shafts between the reduction stages

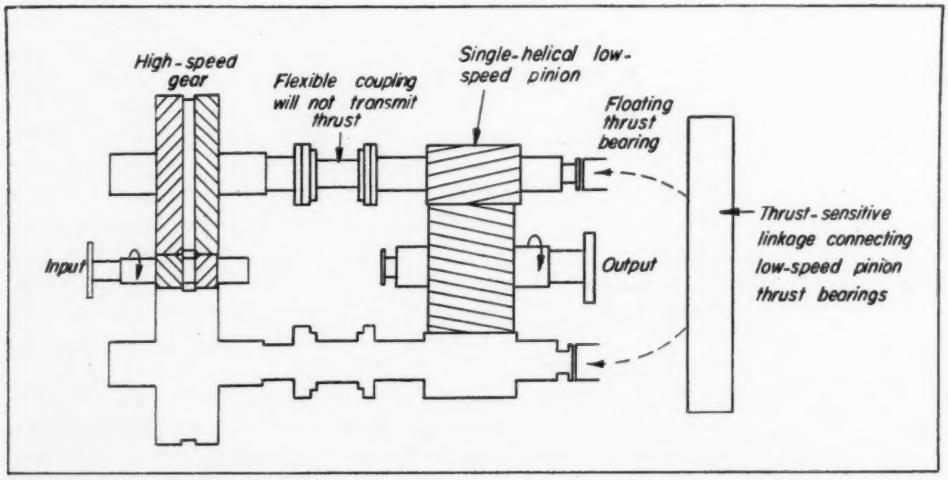
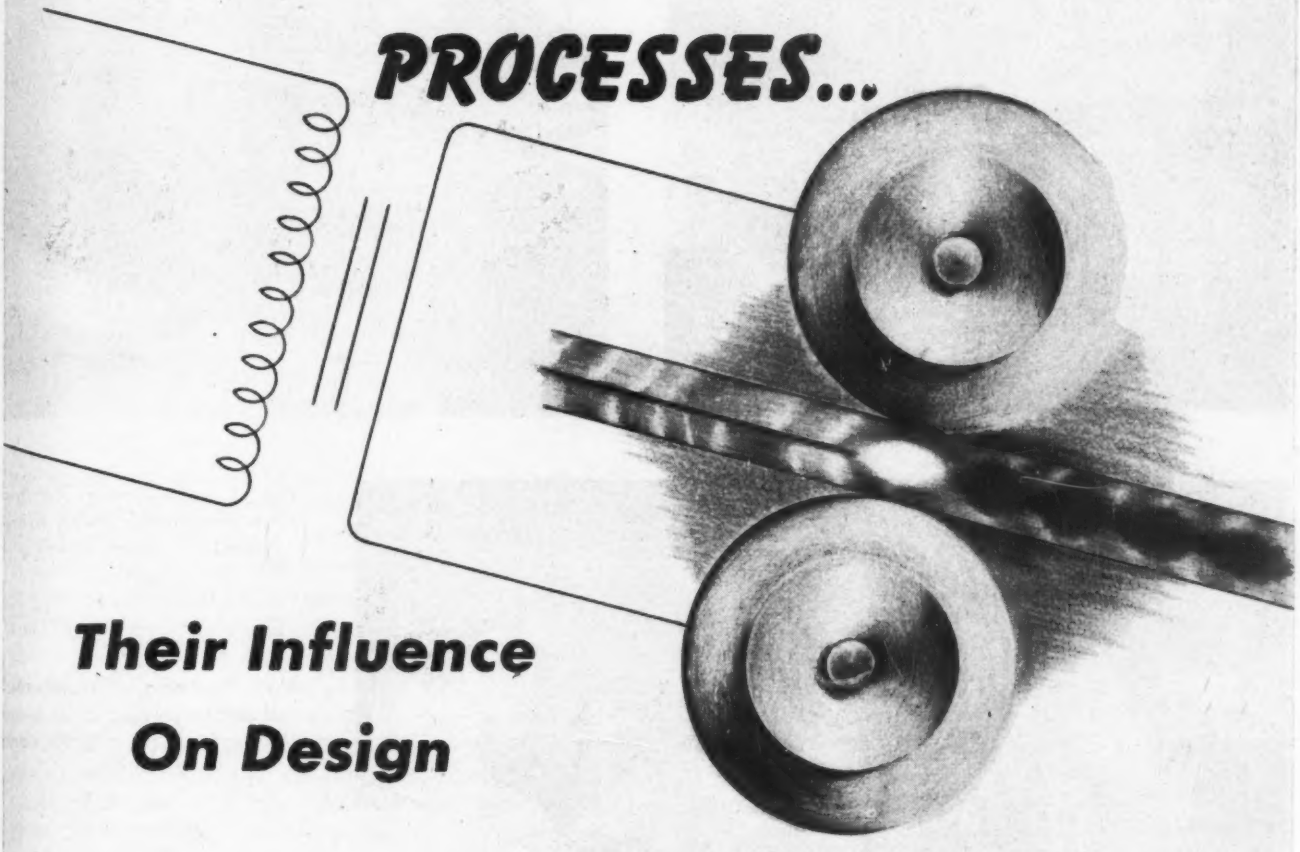


Fig. 10—Left—When single-helical gears are employed, a simple thrust-sensitive linkage, connecting the floating thrust bearings of the intermediate pinions, serves to equalize loading on gears automatically

PRODUCTION PROCESSES...



Their Influence On Design

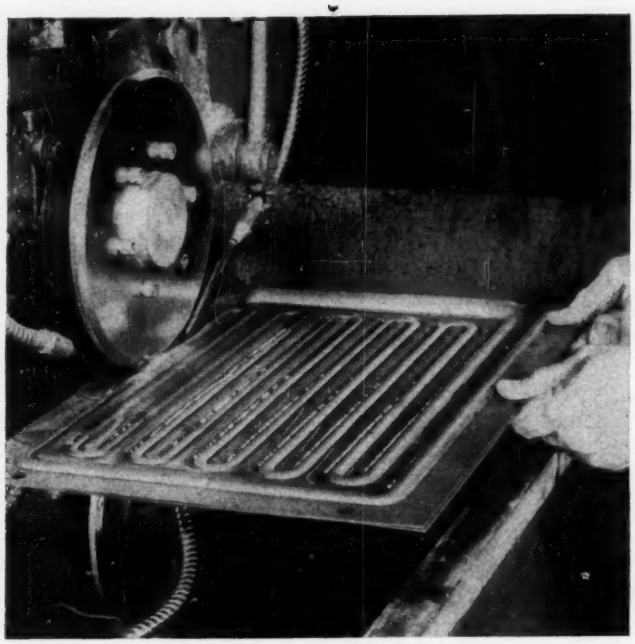
By Roger W. Bolz
Associate Editor, Machine Design

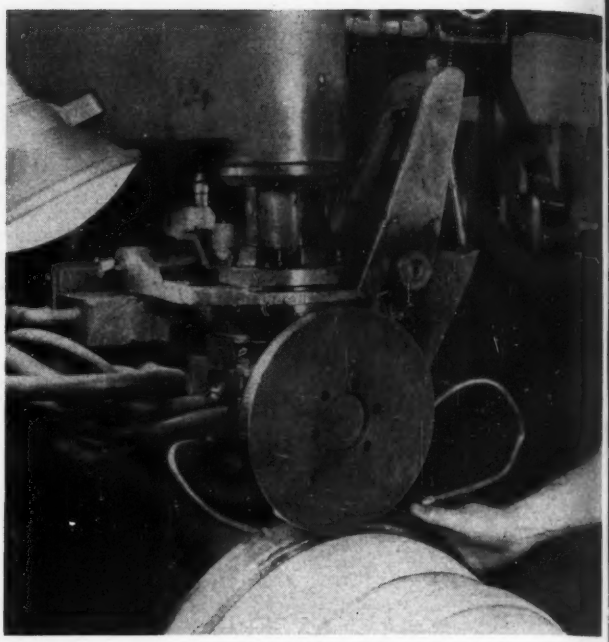
Part XXXIX–Seam Welding

IN PRINCIPLE, seam welding is accomplished in the same manner as in single-spot welding. Combined use of heat, generated by resistance of the interfaces of the metal to the passage of an electric current, and applied pressure to forge together the fused metal produces each weld. Application of pressure and current in seam welding, however, is by means of rotating, wheel-like electrodes, *Fig.* 1.

Unlike spot welding where the electrodes are opened and closed to produce each weld, work to be seam welded is passed between the wheels which remain in contact during the entire operation. When the circuit is closed, the current produces a weld at the point of pressure between the wheels. The wheels continue to rotate and after a preset off-time interval, another surge of current produces the next weld, and so on.

When the work is passed between the electrodes so that the line of weld is at right angles to the throat of the welder, the process is usually termed *transverse* or *circumferential* seam welding, *Fig.* 2. However, when the work is processed so that the weld seam is parallel to or





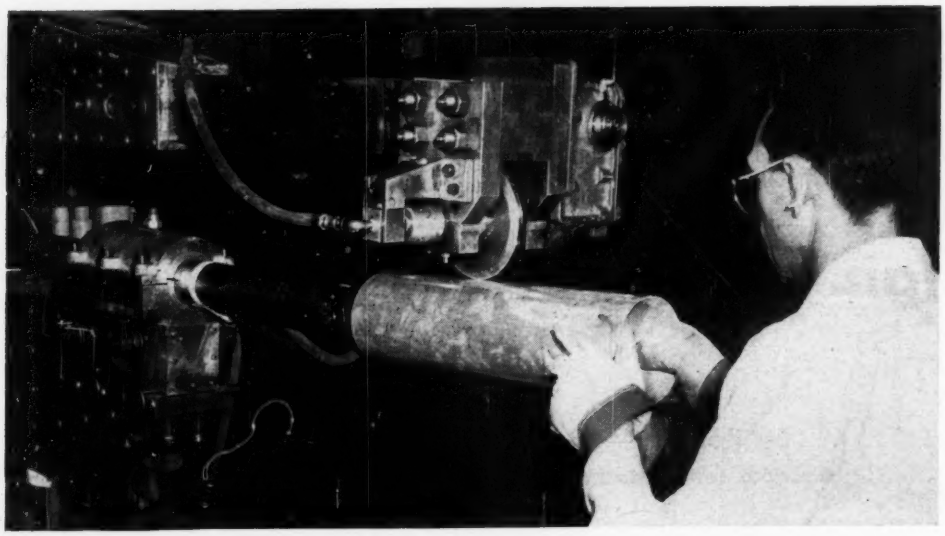


Fig. 1—Left—Above—Refrigerator evaporator being fabricated by seam welding

Fig. 2—Above—Circumferential seam welding of a large flange to a jet engine cone assembly

Fig. 3 — Left — Longitudinal seam welding of aluminum aircraft duct. Tube is close to minimum diameter

into the throat of the machine, it is normally termed *longitudinal* seam welding, *Fig.* 3.

Several types of seam welds can be produced and selection naturally depends upon the application. Normal practice in seam welding is to produce *pressure-tight* joints by adjusting wheel speed and off-time to obtain a succession of overlapping spot welds, *Fig.* 4. Actual fusion of the metal must overlap and the required number of spots per inch varies with metal thickness, TABLE I. Occasionally it is desirable to have a spaced series of spots for joints not necessarily pressure-tight. Known as *roll-spot* welding, *Fig.* 4, such a series of closely spaced spots are produced by adjusting wheel speed and off-time so that individual welds are produced as the work is fed between the wheels. This type of seam weld is capable of great speed, certain machines producing for instance 600 spots per minute spaced ½-inch apart on 22-gage mild steel. Third type of joint, known as the *mash-seam* welded joint, is produced by reducing the normal joint overlap to approximately 1½ times the gage so that the two pieces are forged into each other as the welding progresses, *Fig.* 4. The resultant joint thickness is slightly greater than one sheet thickness and closely resembles a butt joint. Limitations of this joint are requirements for tack or spot welding, clamping, or holding in a jig to retain part dimensions and difficulty in obtaining satisfactory joints in any but the lighter gages.

Standard or special machines to produce any of the types of joints by the foregoing seam-welding methods may be classified generally as: (1) Circular, (2) longitudinal, or (3) universal. Thus the machine is classified primarily by the method whereby the work is fed into the machine as discussed previously.

CIRCULAR SEAM WELDERS: In all probability the most widely used machine, the circular seam welder, is designed to produce transverse or circumferential seam welds. Because the work is fed into the machine at right angles to the welder throat or arms, length of continuous seam is almost unlimited, *Fig.* 5. As can be seen, these machines are used primarily

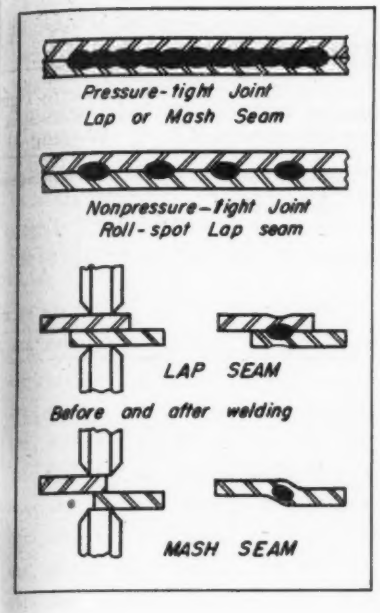


Fig. 4—Left — Types of seam welded joints utilized

Fig. 5—Right — Roll-spot welding a reinforced doubler to aluminum channel

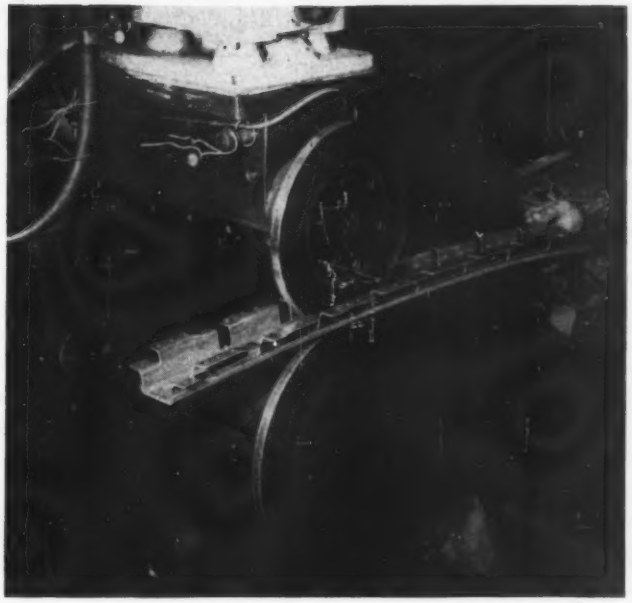


Fig. 6—Below—Seamwelding a flexible section for the jet tailpipe of a B-45 bomber

for making presure-tight and roll-spot circular or end seams on cylindrical, *Fig.* 6, as well as single or multiple long, straight or curved seams. Standard RWMA machine sizes are shown in TABLE II.

LONGITUDINAL SEAM WELDERS: For producing longitudinal seams, the longitudinal welder operates with the work being fed into the machine throat, *Fig.* 7. These machines are adapted primarily for making side or straight seams on cylindrical or tubular work of varying cross section. Generally, this particular seam welding method is limited to rather short, straight seams; as a rule approximately 6 inches less than the available throat depth of the particular machine, TABLE II.

UNIVERSAL SEAM WELDER: Welders which combine the basic features of both longitudinal and circular machines are generally termed universal seam welders. These are designed so as to permit the upper welding head to be swiveled 90-degrees to either position and have a replaceable lower arm with welding units to match. Welds of either the circular or the longitudinal type can be readily made

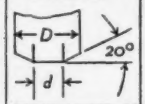
SPECIAL MACHINES: In addition to the regular run of standard machines, a variety of special or semi-special units are widely used. These include: (1) Machines with traveling wheels to allow the work to be clamped stationary while the seam is completed; (2) machines with a traveling platen to allow the work to be passed under the roll to complete the seams; (3) machines with multiple rolls to complete more than one seam simultaneously, *Fig.* 8; (4) machines designed for portable seam welding of unusual parts, *Fig.* 9; and (5) machines specially arranged to facilitate difficult operations in mass production, *Fig.* 10.

PRODUCTION: In conventional machines the wheel or roll drive may take several forms: The upper or lower rolls may be direct gear driven or powered by a friction drive. The upper roll may be positively gear driven with the lower roll idling or, in the case of circular work, the lower may be driven with upper roll idling. Friction drive with knurled drive rolls is the only method that permits power application to both rolls and therefore is most widely used. Where space requirements do not permit driving the lower roll only the upper is driven. Tin plate, terne plate, galvanized steel, and other coated steels should be welded on knurl driven machines inasmuch as the knurled drive rolls continuously break up the surface of the welding wheels thereby removing pieces of the coating, picked up during passage over the sheet.

Speed of passage of the work through the wheels normally ranges from 20 to 150 inches per minute, although higher speeds up to about 50 feet per minute

TABLE I†

Seam Welding Low-Carbon Steel With Interrupted Current[1]

Data Common To All Welding Speeds								Pressure-Tight Joint														Nonpressure-Tight Joint Roll-Spot Welding			
								Maximum Welding Speeds Recommended					Average Welding Speeds Recommended					Minimum Welding Speeds Recommended								
Thickness of work pieces	Electrode[2]			Electrode Force		Min. Overlap L		Heat Time	Cool Time	Welding Speed	Welds per Inch	Welding Current	Heat Time	Cool Time	Welding Speed	Welds per Inch	Welding Current	Heat Time	Cool Time	Welding Speed	Welds per Inch	Welding Current	Heat Time	Cool Time	Welding Speed	
																									
(in.)	Min. d (in.)	Nor. d (in.)	Min. D (in.)	Min. (lb)	Normal (lb)	Use with min. d (in.)	Use with nor. d (in.)	(c)[3]	(c)	(In. per min)		(amp)	(c)	(c)	(In. per min)		(amp)	(c)	(c)	(In. per min)		(amp)	(c)	(c)	(In. per min) Max.	Avg.
0.010	1/8	5/16	5/8	400	400	1/4	5/8	1	1	116	15.5	11,500	2	1	80	15	8,000	2	3	49	14.5	7,500	2	*	480	200
0.021	1/32	5/16	5/8	460	550	5/16	7/16	2	1	107	11	12,500	2	2	75	12	11,000	3	3	45	13	9,000	2	*	425	185
0.031	1/32	1/4	1/2	530	700	5/16	1/2	2	1	103	11.5	15,000	3	2	72	10	13,000	2	4	42	14	12,000	3	*	370	180
0.040	5/32	1/4	1/2	600	900	3/8	1/2	2	2	98	9	18,300	3	3	67	9	15,000	2	4	39	15.5	13,500	3	*	310	170
0.050	7/32	5/16	1/2	700	1050	7/16	9/16	2	2	95	9.5	20,000	4	3	65	8	16,500	4	4	37	12	14,000	4	*	250	180
0.062	7/32	5/16	1/2	750	1200	7/16	5/8	3	1	91	10	21,000	4	4	63	7	17,500	4	4	36	12.5	15,400	4	*	190	150
0.078	7/32	3/8	5/8	980	1500	1/2	11/16	3	1	85	10.5	22,000	6	5	55	6	19,000	6	6	30	10	16,000	5	*	170	140
0.094	1/4	7/16	5/8	1010	1700	1/2	3/4	4	2	80	7.5	23,000	7	6	50	5.5	20,000	6	6	27	11	17,000	6	*	160	120
0.109	9/32	1/2	3/4	1120	1950	9/16	13/16	4	2	75	8	25,000	9	6	48	5	21,000	6	6	25	12	18,500	6	*	150	100
0.125	5/16	1/2	3/4	1240	2200	5/8	7/8	4	2	70	8.5	27,500	11	7	45	4.5	22,000	6	6	23	13	21,000	6	*	140	90

† Courtesy Taylor-Winfield Corp.

[1] Low-carbon steel is hot rolled, pickled, and lightly oiled with an ultimate strength of 42,000 to 45,000 psi similar to SAE 1005—SAE 1010. Surface of material lightly oiled but free from grease, scale or dirt.

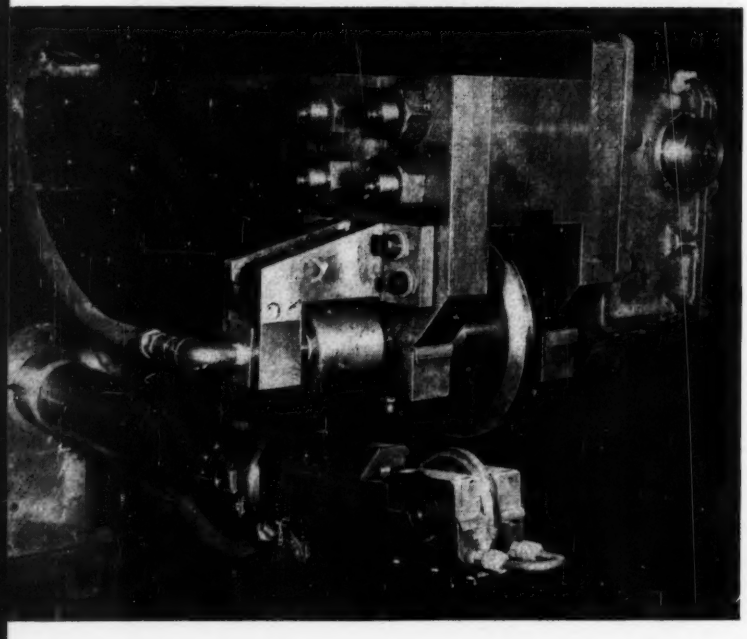
[2] Electrode material is RWMA Class 2 alloy. Radius crowned electrodes work satisfactorily in seam-welding low-carbon steels also.

[3] Heat and cool times are given in cycles of 60-cycle frequency.

have been used with continuous application of current.

LIMITATIONS: Seamwelding is a fast and economical method of joining sheet material together for gas or liquid-tight joints but unlike spot welding is mainly limited to the lighter gages. The general range of thicknesses normally considered in the seam welding category is from about 0.010-inch to 1/8-inch. Although it is possible to seam weld gages to as thin as 0.002 to 0.003-inch, this is seldom done since it requires special equipment. Top limit of 1/8-inch is seldom exceeded as above this gage production equipment becomes too costly and output too slow.

Fig. 7—Longitudinal seam welder showing roll arrangement



DESIGN: Seam welding, like spot welding, is most readily applied and successfully used if the various aspects and limitations incident to the process are considered. By proper design practice maximum economy likewise will be assured.

As previously noted, seam welding is somewhat limited as to the range of sheet thicknesses which can be welded. Generally two low-carbon sheets from 0.010-inch to 1/8-inch define the maximum and minimum limits. In stainless, sheets from about 0.006-inch up to 1/8-inch can be handled on regular machines.

Pressure-tight joints require a minimum number of spots per inch to obtain continuous fusion. Thus approximately 12 per inch are required for 0.010-inch stock ranging down to about 5 for 1/8-inch material. For a given material and thickness combination there is a definite maximum welding speed determined by the maximum possible rate of heat input which is limited by surface burning of the material and rate at which the welds solidify under wheel pressure. Maximum speeds and welds per inch are given in TABLE I.

Proper Joint Overlap Required

Also given in TABLE I are the recommended joint overlaps for various gages of low-carbon steel. Parts to be lapped or flanged and welded require sufficient overlap or flange width to prevent burning of the edges or splattering. Two overlap dimensions are shown in the table but the larger is preferred and should be employed wherever possible. The smaller dimension only provides *reasonably* good results and

it usually is necessary to use fixtures to assure centering of the weld on the narrow flange. For large assemblies the contacting overlap should be increased by 30 per cent to insure sound joints and facilitate guiding.

Proper overlap and fitup is of primary importance in seam welding. Where design or features of an assembly are such as to present difficulty in controlling the overlap in the welding operation, the necessary lap should be fixed by a few preliminary spot welds. Fitup of joints to be welded should be good and seams should require no electrode force to create a good, close fit during the weld. Circular seam welds are particularly critical in this respect and wherever possible a press fit or similar means for insuring intimate contact in the welding zone should be used where best quality welds are required.

Where circular welds, *Fig.* 6, are concerned, the smallest inside diameter which can be handled is about 4 inches. This allows a suitable electrode diameter and reasonable service life. Smaller diameters can be welded but require special equipment and technique. Where cross sections of such welds are other than round, the minimum internal radius allowable at corners is 2 inches. Preferable radius is 3 inches on steel and 6 inches for aluminum. On large square sections, for instance, use of 2-inch radius corners slows down the straight welds to the speed required in traversing the small corner radius and in such cases it is preferred to design-in as large a radius as possible to speed output. Position of such welds in relation to the part length is also limited as is the material gage, *Fig.* 11. Design of the joint for dished or flat ends as well as side or centered joints must be such as to allow access to the rolls, *Fig.* 12. Where part design does not allow roll clearance, special electrodes can sometimes be used, *Fig.* 13, but production must warrant such special attachments.

Longitudinal seams on cylindrical or tubular parts of varying cross section are limited, as noted previously, by the throat of the welder available. Length of seam which can be produced should be restricted to about 6 inches less than the available throat length. Where both ends are open the possible seam length can be doubled. As a rule, outstanding seam

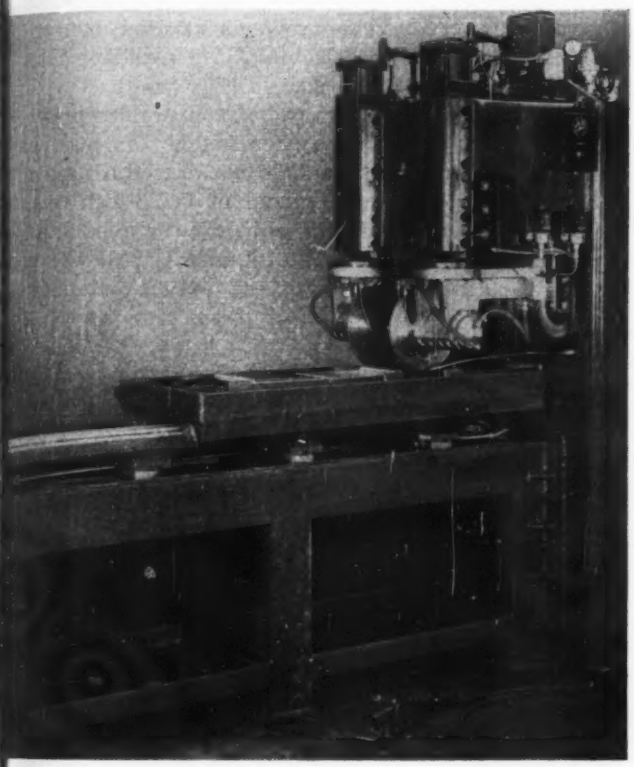


Fig. 8 — Above — Dual-wheel seam welder for fabricating refrigerator evaporators. Operation with shuttle table is automatic

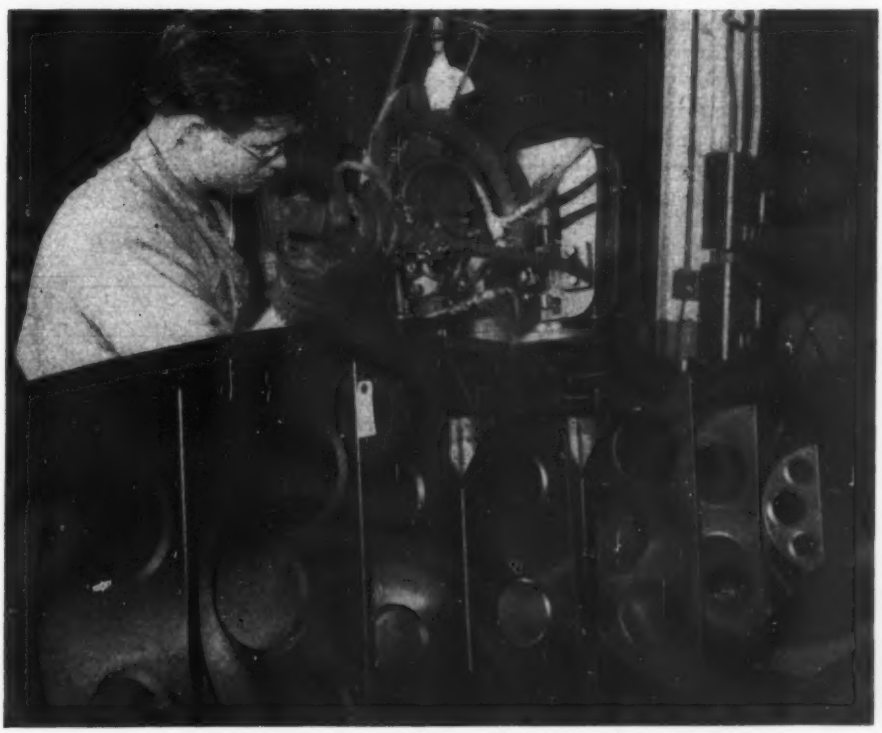


Fig. 9 — Right — Special portable seam welder used to join the pan and half shell of aircraft drop tank

welds on tubular parts prove the most economical design for small quantities while overlapped seam welds are more economical in large quantities.

In welding flanged parts such as tanks where the weld must turn along as external radius, the minimum external radius advisable is 2 inches for 22 through 18-gage steel (0.030 through 0.050-inch thick), 2¼ inches for 16-gage (0.062-inch) steel, 2½ inches for 14 gage (0.078-inch) steel, and 3 inches for 13-gage through 11-gage steel (0.094-inch through 0.125-inch). Minimum radius in aluminum is 3 inches. On concave radii where the sides are normal to the flange, the inside radius should be no less than 10 inches in aluminum and 6 inches in steel. When the sides form part of a circle with radius not exceeding 4 inches, the welding radius can be reduced to a minimum of 6 inches in aluminum and 3 inches in steel, *Fig.* 14.

TABLE II

Specifications for Standard Seam Welders

Size. No.	Rating (kva)	Throat Depth (in.)
1	50—75	18—24—30
2	100—150—200	18—30—42
3	250—400	24—36—48

The good and poor design practices shown in *Fig.* 11 of the preceding article on spot welding apply largely in seam welding also. Typical wheel diameters may range from a minimum of 2 inches to as much as 24 inches with widths shown in TABLE I. Joint design should allow for use of standard setups wherever possible.

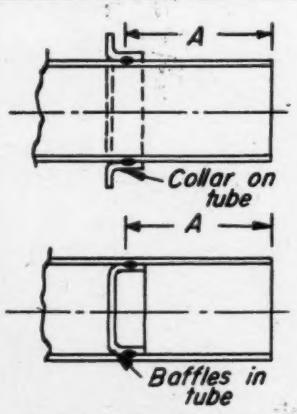
Width of weld at the centerline between the work pieces ranges from 1.5 to 3 times the gage of the metal being welded. The higher ratio of weld width to sheet thickness occurs in welding the thinner gages and the ratio decreases with increase in gage. The practical width of wheel face is limited to a minimum of about ⅛ to 3/16-inch by excessive wear; hence the higher ratios with the lighter gages.

In completing the design of a seam-welded part, much as in spot welding, it is essential that complete data regarding the joints be included. In supplying such essential information, use of the recommended standard welding symbols developed by the American Welding Society is considered good practice. These symbols are summarized in the legend shown in *Fig.* 13 of Part XXXVIII in this series which appeared in the November issue.

Where obstructions which break up the continuity of the welded seam must be present in a part, these can often be straddled by the use of notched wheels. If the obstruction is on one side only, a single notched wheel and a round wheel can be used or in the case of obstructions on both sides, two geared notched wheels are used. Start of the weld is made with the notch close to the obstruction and the seam is made by rolling away from the obstruction with the wheel. The obvious production problems created by such designs indicate the desirability for avoiding them whenever possible to do so.

Although gage of sheets to be seam welded preferably should be equal, such need not be the case. In aluminum unequal gages should not exceed a ratio of 2 to 1 with a maximum pileup of 0.128-inch total. In low-carbon steel the ratio should not exceed 3 to 1 or a maximum pileup of 0.150-inch. Likewise in stainless, ratio should not exceed 3 to 1 but pileup

Fig. 10—Above—Special revolving head machine for welding a filler well to a gas tank shell. Head revolves on vertical axis to produce a circular seam weld completely surrounding the filler cap



I.D. of Tube (inches)	Steel Max. Gage (inches)	Aluminum Max. Gage (inches)	Dimension 'A' Max. (inches)
4-1/4 to 5	.050	.032	3
5-1/4 to 6	.075	.040	3
6	.075	.040	6
7	.075	.040	10
8	.075	.040	12
9	.075	.040	14
10	.075	.040	16

Fig. 11—Left—Limitations of circular seam welds relative to tube diameter, material gage and location in collar and baffle arrangements

may range to 0.186-inch, maximum. In welding three thicknesses (maximum) the following restrictions apply: (1) Similar alloy combinations, (2) a maximum thickness ratio of 2 to 1, and (3) applications on which indentation can be permitted on both sides.

Where thickness through a finished weld must be a minimum, mash-seam welding is used. A slight overlap of 1 to 1½ times the gage is provided and the pieces must be held by clamps or spot welding. Spot weld tacking is preferred and for proper holding will vary from 1½-inch spacings on 0.020-inch stock to 3-inch spacings on 0.078-inch stock. From mash welding the resulting seam is about 10 to 25 per cent thicker than the original gage. Mash-seam welding is limited to low-carbon and coated steels of 14-gage (0.078-inch) or less and should not be used unless absolutely necessary.

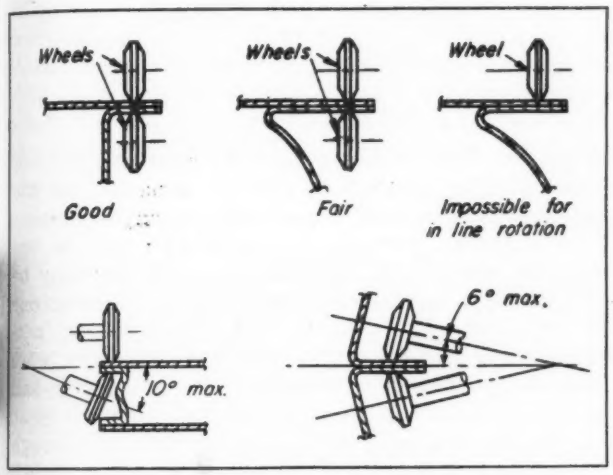
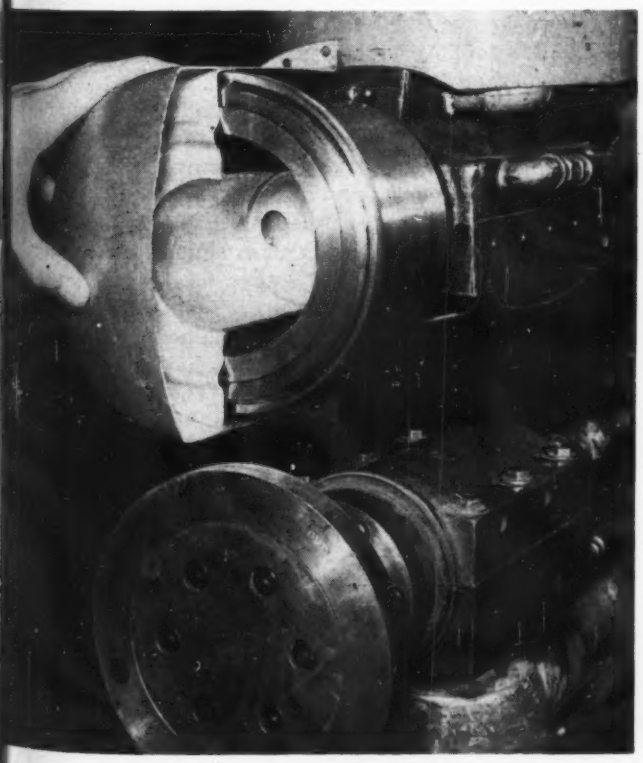


Fig. 12—Above—Design of joints for dished or flat ends to allow access to rolls

Fig. 13—Below—Special hollow electrode developed for seam welding a special hydraulic reservoir



MATERIALS: Generally speaking, the data covering selection of materials for spot welding, included in the previous article in this series, also applies to seam welding. Low-carbon steel should be free from any rust, scale, paint, grease or oil. Also, if possible, the material should not have any coating where highest quality welds are desired although steels with zinc, tin or lead-coated surfaces can be welded.

Comparative weldability of various metals covered in TABLE III of the previous article also applies as does the weldability of dissimilar metals, TABLE IV.

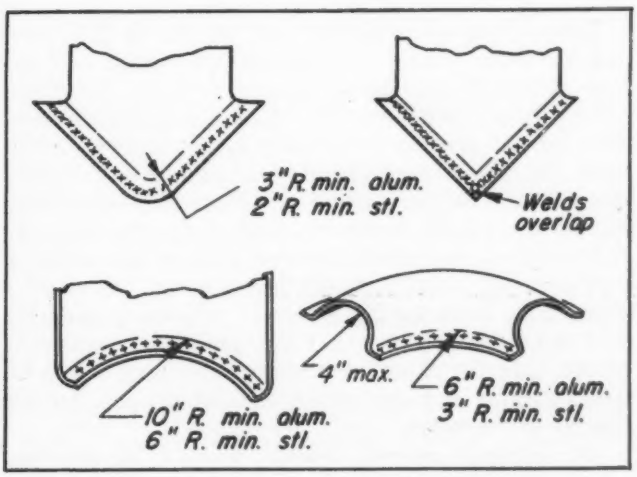
TOLERANCES: As with spot welding, seam welding has little effect on part tolerances, specific working dimensions of welded parts being dependent upon the method used in fabricating the various components. Where precise locational tolerances are necessary, fixtures can be employed, *Fig.* 9, or the parts can be tack welded prior to seam welding.

With any properly made seam weld the wheels leave depressions which normally can not be prevented. Where marring of the surface must be prevented, series welds using a wide wheel or shunting bar can be employed as with spot welding.

Collaboration of the following organizations in the preparation of this article is acknowledged with much appreciation:

Federal Machine & Welder Co. ..... Warren, Ohio
General Electric Co.
(*Figs.* 1 and 2) ..... Schenectady, N. Y.
Lockheed Aircraft Corp.
(*Fig.* 3, 7, 9, 10, 11, 12, 13 and 14) .... Burbank, Calif.
P. R. Mallory & Co. ..... Indianapolis, Ind.
North American Aviation Inc.
(*Figs.* 5 and 6) ..... Los Angeles, Calif.
Progressive Welder Co. (*Fig.* 8) ..... Detroit, Mich.
Resistance Welder Manufacturer's Association
(Table II) ..... Philadelphia, Pa.
Taylor Winfield Corp.
(*Fig.* 4 and Table I) ..... Warren, Ohio
Westinghouse Electric Corp. ..... Pittsburgh, Pa.

Fig. 14—External radii on flanged parts are limited to the minimum values shown. Where internal radii on flanged parts are used, larger radii are necessary to permit access to the wheel



Superprecision Ball

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Part II—Selection for Grade of Accuracy and Speed

FOR MANY years, it was customary in the bearing industry to specify tolerances of ball bearings as "SAE Standard" or "One-half SAE Standard". The SAE Standard for boundary dimensions and errors of eccentricity is used for the vast majority of applications where no great degree of precision is required. Where more accurate bearings were required, it was previously the practice to specify one-half the tolerances allowed in the SAE Standard. However, since other errors such as parallelism and squareness between grooves and outside surfaces were not included in the SAE Standard, it was finally decided that such errors should be recognized and limited by a set of specifications.

During the 1930's, the ABEC Dimensional and Tolerance Standards were adopted and these standards have become generally accepted both in this country and abroad. These finally revised tolerance standards are as follows:

ABEC-1. Standard
ABEC-3. Selected, in which the tolerances are closer than Standard on bore, outside diameter and eccentricity of inner and outer ring
ABEC-5. Precision, in which the tolerances are closer than ABEC-3 on bore, outside diameter and eccentricities and include tolerances on errors of squareness and parallelism
ABEC-7. Superprecision, in which the tolerances are closer than those of ABEC-5.

TABLE I shows a partial size listing of the permissible errors in the four grades of bearings. These four standards pertain to annular bearings only and not to thrust bearings. However, the present standards do not specify differences in accuracies of the balls for the various grades. In many cases, variations in size and roundness of the balls considerably affects the results obtained on precision applications. Individual users have demanded greater and greater ball accuracy so that today bearings are being produced with balls having errors of sphericity and size held within 0.000010-inch. The published standards of the industry show permissible sphericity errors of 0.000025-inch and size errors of 0.000050-inch for all grades of bearings.

Of particular interest to many users is the growing realization that the magnitude of apparent errors of eccentricity in bearings is not as important as the repetition of errors of eccentricity during a number of revolutions. Perhaps this point can best be appreciated when it is realized that a bearing may be accepted with an error of eccentricity during one revolution of the inner or outer ring within permissible limits but would be rejected if the test were repeated for several revolutions. Since angular ball travel is always less than the angular inner or outer ring travel, new balls are constantly passing through the area of support and errors of groove and ball roundness and size result in nonrepetitive runout errors not only radially but axially as well.

Spindle extensions can be ground true and round on their bearings only to the degree of accuracy permitted by the sum of the roundness errors of the grooves and the roundness and size errors of the balls within the bearings. Preloading of the bearings, however, compensates for some of these errors due to elasticity of the materials comprising the balls, rings and their mountings.

While ball bearings are mass produced, demands for high accuracy inevitably result in cost increases in proportion to the degree of accuracy demanded. Better accuracy not only demands more skill upon the part of machine operators and inspectors but constant attention to the proper maintenance of accuracy of the machines, tools and gages used. Inevitably, improvements in bearing accuracy demand improvements in the accuracy of the machine spindles and their bearings used in the production of bearings. Accuracy of bearings produced is therefore closely dependent on the accuracy of the bearings of the machine itself and further improvements are dependent entirely on the machine-and-spindle bearing combination.

Selection For High Speed

The earliest ball bearings were manufactured with a full complement of balls with no attempt to space the balls and reduce their adjacent rubbing action. As the art of bearing manufacture progressed, separators or retainers of metal were applied with the

Bearings

result that operating speeds could be materially increased as the effective rubbing speed was cut in half. As the demands for speed grew, it was soon recognized that the best results were obtained with fully machined metallic ball retainers piloted on the lands of the inner or outer ring, thus achieving much closer control of eccentric running and relieving the balls of the weight of the retainer. However, it was noted that rather rapid wear developed under some operating conditions between the inner ring lands and the retainer piloting surfaces. Metallic seizure is apt to develop with indifferent lubrication, and eccentric wear often develops due to slight errors of eccentricity in the retainer itself, resulting in out of balance running.

Superior results were found to be obtained with lightweight plastic materials. Phenolic laminated plastics have proved to be the best solution, although permissible operating temperatures are limited by the loss of strength which takes place in the vicinity of 275 F. Since the specific gravity is low (1.1—1.3) compared with metallic materials, unbalanced forces due to machining errors will be low as will be the tensile hoop stresses. These factors outweigh the apparent disadvantage of a low-strength material. In addition, repeated tests have proved the nonseizing characteristics of phenolic laminates where there is an occasional lack of lubrication. While only a very small percentage of oil is absorbed into the material itself this small quantity of lubricant is sufficient in

TABLE I

Comparison of ABEC Tolerance Grades for Assembled Bearings

(All tolerances and errors in 0.0001-inch units)

Inner Rings

Bore (mm)		Bore Tolerance*				Eccentricity (Max. error)				Parallelism of Sides (Max. error)		Side Runout With Bore (Max. error)		Groove Parallelism With Sides (Max. error)	
		ABEC Class				ABEC Class				ABEC Class		ABEC Class		ABEC Class	
over	incl	1	3	5	7	1	3	5	7	5	7	5	7	5	7
0	9	+1 −4	+1 −3	+0 −2	+0 −1½	3	2	2	1	2	1	3	1	3	1
9	18	+1 −4	+1 −3	+0 −2	+0 −1½	4	3	2	1	2	1	3	1	3	1
18	30	+1 −5	+1 −3	+0 −2	+0 −1½	5	3	2	1½	2	1	3	1½	3	1½
30	50	+2 −7	+2 −5	+0 −2	+0 −2	6	4	2	1½	2	1	3	1½	3	1½
50	80	+2 −8	+2 −6	+0 −3	+0 −2	8	4	2	1½	2	1½	3	2	3	1½
80	120	+3 −11	+3 −8	+0 −3	+0 −2½	10	5	3	2	3	1½	3	2	4	2
120	180	+3 −13	+3 −9	+0 −4	+0 −3	12	6	3	3	3	2	4	3	4	3
180	250	+4 −16	+4 −11	+0 −5	+0 −4	16	8	4	3	4	2	4	3	5	3

Outer Rings

O. D. (mm)		O.D. Tolerance*				Eccentricity (Max. error)				Parallelism of Sides (Max. error)		Side Runout With O.D. (Max. error)		Groove Parallelism With Sides (Max. error)	
		ABEC Class				ABEC Class				ABEC Class		ABEC Class		ABEC Class	
over	incl	1	3	5	7	1	3	5	7	5	7	5	7	5	7
0	18	+1 −5	+1 −4	+0 −2	−0 −2	6	4	2	2	2	1	3	1½	3	2
18	30	+1 −5	+1 −4	+0 −2	+0 −2	6	4	2	2	2	1	3	1½	3	2
30	50	+2 −7	+2 −5	+0 −2	+0 −2	8	4	2	2	2	1	3	1½	3	2
50	80	+2 −7	+2 −6	+0 −3	+0 −2	10	5	3	2	2	1	3	1½	4	2
80	120	+3 −9	+3 −7	+0 −3	+0 −3	14	7	4	2	3	2	3	2	5	2
120	150	+3 −11	+3 −8	+0 −4	+0 −4	16	8	4	3	3	2	4	2	5	3
150	180	+3 −13	+3 −9	+0 −5	+0 −4	18	9	5	3	3	2	4	2	6	3
180	250	+4 −16	+4 −11	+0 −5	+0 −4	20	10	5	4	4	3	4	3	6	4
250	315	+4 −18	+4 −12	+0 −5	+0 −5	24	12	6	4	5	3	5	3	7	4

* Extreme variations permitted with two-point diameter measurement and include errors of taper and roundness. Errors of eccentricity, side parallelism, side runout, squareness and groove parallelism are maximum errors (total indicator readings) permitted during one revolution of the measured parts.

many cases to prevent wear.

Best results have been achieved with the cotton fabric bases, for the most part made in the form of rolled tubing. Originally, flat laminated stock generally was required for the two-piece retainers used in deep-groove bearings where screwed or riveted fastenings are required. Recent developments, however, indicate equal strength and reliability from rolled tubing for both single and two-piece retainers. Under no conditions should abrasive materials be considered as laminates or fillers for the manufacture of ball retainers.

Critical factors which affect ball bearings running at high speeds are as follows:

1. Geometrical trueness and uniformity of size and fitting of the component parts of the bearings. This to insure freedom from undue stresses and to promote maximum true rolling action
2. Accuracy of mountings and seatings for the bearings. This avoids excessively tight or loose fitting and promotes good alignment as well as insuring freedom from undue stresses and promoting true rolling motion of the bearings
3. Ability of the ball retainer to endure rubbing at high speeds
4. Ability of the mounting and/or lubrication arrangement to dissipate heat
5. Ability of the seals to exclude foreign matter and other damaging substances
6. Quantity and type of lubricant most suitable for the application
7. Centrifugal forces developed by the balls against the outer ring raceway groove.

At low speeds the centrifugal forces are of small order and may be neglected as the inner ring raceway groove is the critical member from the fatigue standpoint. As speeds increase the load due to centrifugal forces exerted on the balls becomes more pronounced. The outer ring of the bearing then becomes the critical member and a point is finally reached when all of the load capacity of the bearing is consumed by the centrifugal force loading on the outer ring and further increases in speed may be tolerated only with a reduction in bearing life.

As far as limiting speeds for the various types of bearings are concerned, this is a difficult subject to cover as so many variables are present in the wide variety of possible installations. Of interest however, is the fact that on a number of applications bearings have been run successfully at speeds beyond the expectations of the bearing manufacturer. *Under ideal conditions* the speed factors shown in TABLE II are possible with light loads.

Under ideal conditions single-row radial bearings with metallic retainers have been operated in rare instances at speeds approaching those shown in TABLE II. However, the usual pressed steel or brass retainer is not inherently suitable for high-speed operation. Machined retainers of brass and bronze generally perform better than pressed retainers. From knowledge gained so far it seems that the ideal retainer material is phenolic composition unless temperatures exceed 275 F.

Single-row loading groove and double-row radial and angular-contact bearings generally can only be equipped with pressed steel retainers and are therefore limited to about the same speed restrictions as deep-groove bearings with pressed steel retainers. However, the inherent disadvantages of these types, as previously discussed, generally limit the possibility of success of these bearings at high speed.

Single-row, angular-contact bearings with high contact angles are furnished with various types of retainers including pressed steel, machined brass or bronze and phenolic composition. However, the limit of speed is not necessarily due to the retainer material or construction but rather to permissible factors of heating due to the high contact angles employed. Adequate cooling will make possible higher speeds.

Thrust bearings of the single and double-direction types are not suitable for high-speed operation. As with angular-contact, high-angle type single-row bearings, the factors of ball slippage to a more pronounced extent limit speed performance to a maximum factor of about 300,000 (pitch diameter, mm, × speed, rpm). Summarizing the situation on these miscellaneous types, aside from the question of high-precision characteristics available, the data in TABLE III are presented.

Third part in this series, to appear in the next issue of MACHINE DESIGN, will cover selection of bearings for low friction, for rigidity and load capacity.

TABLE II

High-Speed Precision Grade Bearings

(ABEC-5 and 7)

Bearing Type	Speed Factors*	Type of Lubrication
Angular-contact, separable and nonseparable low-angle, with composition retainer	2,000,000	Air-oil mist Air pressure fed
Angular-contact, separable and nonseparable, low-angle, with composition retainer	1,500,000	Grease
Deep-groove, open and double-shielded, with composition retainer	1,200,000	Grease
Double-row, self-aligning outer raceway, with composition retainer	1,000,000	Oil or grease

* Product of bearing pitch diameter (mm) and speed (rpm)

where pitch diameter $= \left(\frac{\text{Bore diameter}}{2} + \frac{\text{Outside diameter}}{2} \right)$

TABLE III

Miscellaneous Type Bearings

(Selected, not precision quality)

Bearing Type	Speed Factors*	Type of Lubrication
Angular contact, high contact angle	500,000	Oil or grease
Filling notch, loading groove	500,000	Oil or grease
Double row, radial and angular contact	500,000	Oil or grease
Thrust bearings, single and two-direction	300,000	Oil or grease

* Product of bearing pitch diameter (mm) and speed (rpm)

where pitch diameter $= \left(\frac{\text{Bore diameter}}{2} + \frac{\text{Outside diameter}}{2} \right)$

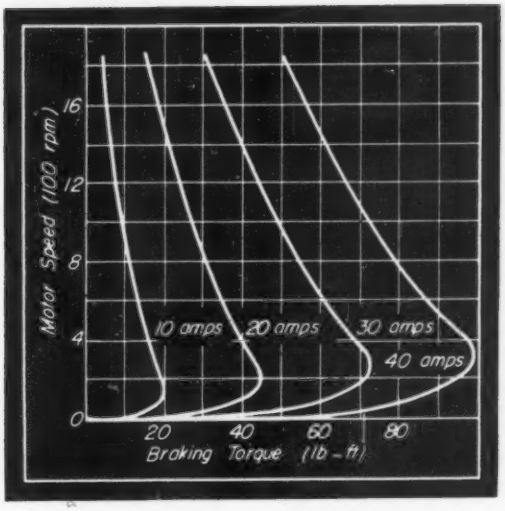

Fig. 1—Curves showing the braking torque developed by a 5-hp four-pole squirrel-cage motor when direct current of the indicated amount flows through any two legs of its 3-phase winding

By F. D. Snyder
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Controlled Braking

. . . of motor-driven machines

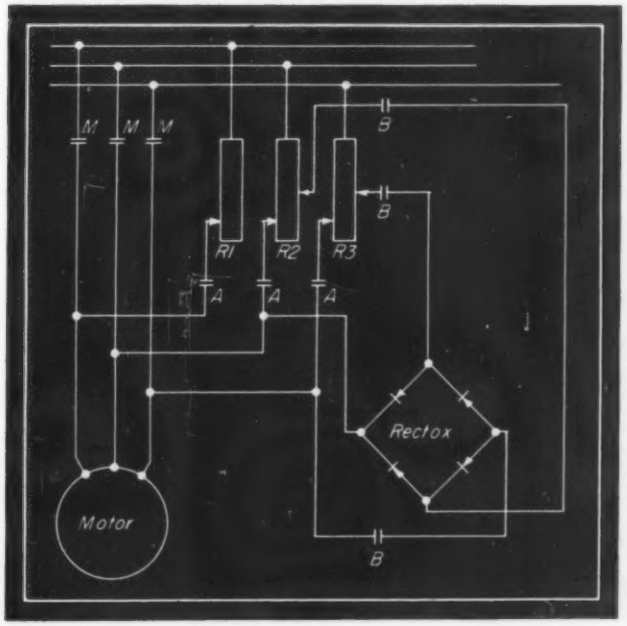
AMONG the exacting functions which electrical controls can perform with improved effectiveness is that of controlled deceleration or braking. The following discussion of typical applications to textile machines will illustrate how the braking requirements of a variety of machines are met in actual practice.

A motor drive for a warp knitting machine should provide low torque for slow and accurate jogging during setup and adjustment periods. Further, it should provide rapid acceleration during regular starting in order to avoid starting marks, together with fast and reliable deceleration to avoid stop marks.

For stopping, solenoid or spring-operated mechanical brakes are not satisfactory because they will not remain in adjustment, and require excessive maintenance. Reversing the power to the motor in conjunction with a zero-speed switch is not satisfactory because it is impossible to decelerate to a complete stop and because the motor may reverse and damage the machine. To achieve successful rapid and positive deceleration a direct-current voltage is applied to two leads of the squirrel cage motor. *Fig.* 1 shows a family of curves for a 5-hp, four-pole, 220-volt, three-phase squirrel-cage motor and indicates the decelerating torque obtainable with different values of direct current. Full load current rating for this motor is 12.5 amp. The value of braking torque varies for different motors but the curves give an indication of what may be expected. The full load torque is 15 lb-ft whereas by this method the braking torque may exceed eight times the full-load running torque. This type of braking is particularly desirable because the braking torque drops off very rapidly as the motor speed approaches zero.

Basic schematic diagram of the control is shown in *Fig.* 2. When stopping the machine, contactor *M* opens and by the use of a suitable timing relay contac-

Fig. 2—Schematic diagram of control circuit for a warp knitting machine, which includes provision of direct current for braking purposes using a rectox rectifier



Based on a paper presented at the summer general meeting of the AIEE in Mexico City.

tor *B* picks up momentarily and applies direct current to the motor through the rectox rectifier and the resistors *R2* and *R3*. The amount of braking can be adjusted by moving the taps on these resistors. With this scheme the motor always comes to a definite stop and cannot reverse even though the direct current should remain on the line. The timing relay is set so that contactor *B* will open just after the machine has stopped.

Stops Within One-Half Revolution

The high-speed dressing reel used in the woolen industry consists essentially of a cylinder about four feet in diameter and from eight to fifteen feet long. A sufficient number of ends or threads for a loom beam are wound onto it, section after section. The sections are built up by pulling the ends from a creel or stand on which are mounted as many individual spools of yarn as there are ends in a section. When a thread breaks or a yardage counter indicates a stop, the reel must stop within one-half revolution in order that a broken thread will not be lost on the reel. As soon as the reel has stopped it must be free to be moved by hand.

Stopping the reel in half a revolution requires such a high torque that it cannot be safely transmitted through the driveshaft. This is solved by the use of two solenoid brakes, one at each end of the reel. Braking torque is also provided by the motor using direct current as previously explained. To initiate stopping, two sets of contactors are dropped out and a timing relay picks up a third contactor which supplies direct current, through a rectox, to the two brake solenoids and two phases of the motor winding, all in series. This contactor opens after the reel has stopped, releasing all braking action and allowing the reel to be rotated by hand. When an end breaks, stopping action is initiated by a multiple unit electronic stop motion relay that will be discussed later.

An improved spooler recently developed has operating requirements similar to the dressing reel except that the stopping period, from the time that an end breaks to dead stop, must be not more than two revolutions of the motor.

Inertia of this machine is not very great and the braking torque available by applying direct current to two leads of the motor primary is found sufficient to stop the machine well within two revolutions. However, too much time is lost before braking action begins with the scheme of opening the line contactor and then picking up the braking contactor.

Use of the arrangement shown in *Fig.* 3 provides satisfactory braking by saving the pickup time of a braking contactor. The stop button, or the protecting device actuated by a broken end, will instantly open contactors *B* and *A* but contactor *M* remains closed. The full-wave rectox rectifier will supply direct current to the motor primary and start the braking action at the instant contactor *B* opens. A timing relay opens contactor *M* and removes all power after the machine has come to rest.

Large shuttle type looms used for the manufacture of paper mill felts, carpets, and other products have, until recently, required a friction clutch type of drive for two main reasons: First, it was the only sufficiently fast way to accelerate the loom to full speed before the shuttle is thrown across the loom the first time; second, it was necessary in order to avoid the flywheel effect of the motor and gears when a protection stop is required. During this protection stop, which is required when a shuttle does not get across

Fig. 3—Diagram of control for a jack spool drive. Dynamic braking stops machine within two revolutions

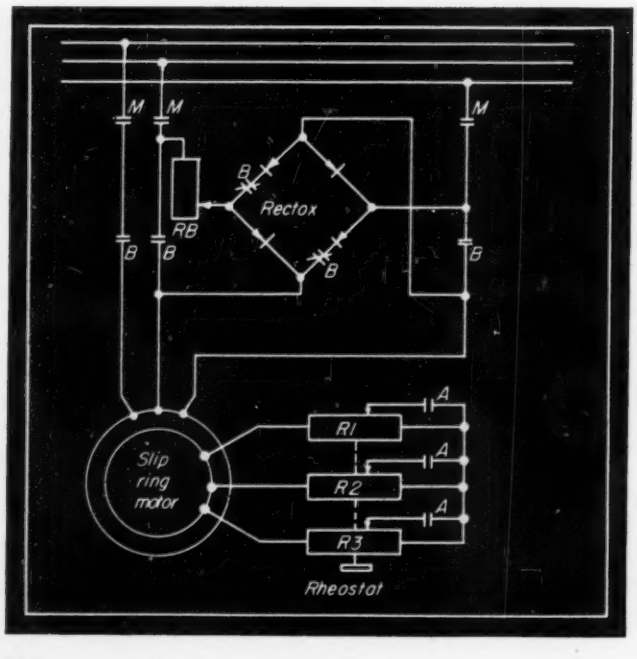
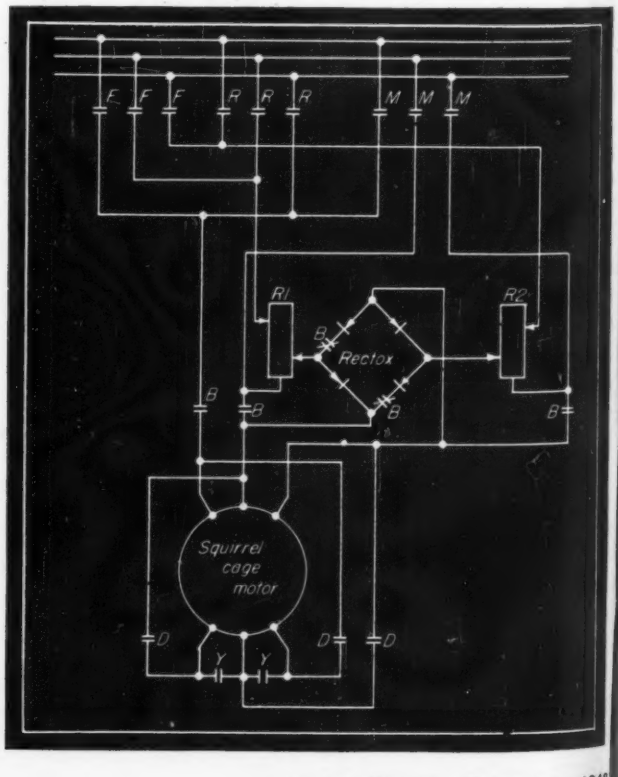


Fig. 4—Control circuit for a large shuttle type loom, which eliminates the need for a mechanical clutch



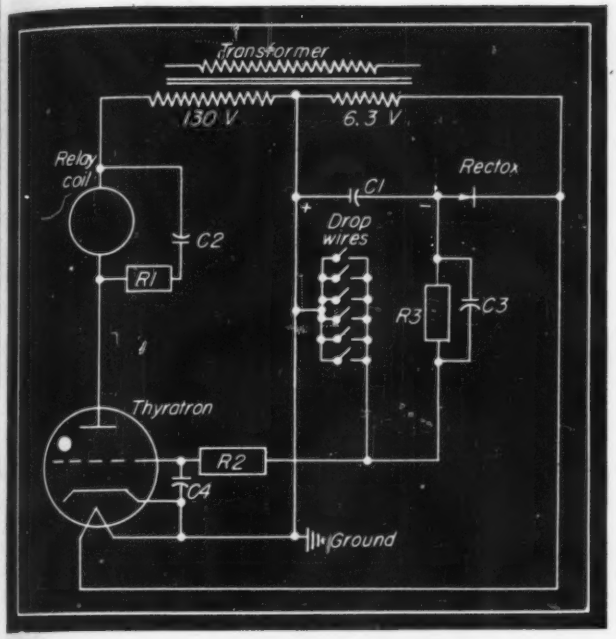


Fig. 5—Schematic diagram of a single unit type of electronic stop motion relay. When thread breaks its drop wire falls, causing grid of thyratron to become positive and permitting the tube to fire

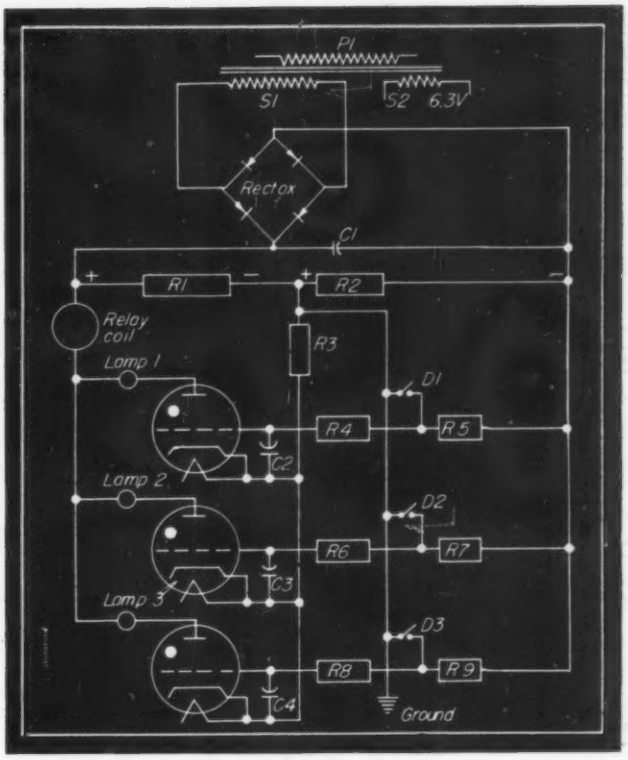


Fig. 6—Multiple unit electronic stop motion relay which shows by lighting a lamp which thread has broken

the shed, the loom is stopped instantly by engaging dogs, or fingers on the moving lay with the stationary breast beam. By releasing the clutch at the same instant, the motor and gears are free and the dogs have to absorb only the energy from the loom itself. This energy on heavy looms is still sufficient to cause loom breakage on protection stops. The clutch has also been desirable to allow the operator to jog the loom accurately during loom fixing or set-up operations.

A drive recently developed will operate these large looms without clutches and will go through the protection stopping without shock to the loom. The basic schematic diagram of the control is shown in *Fig.* 4.

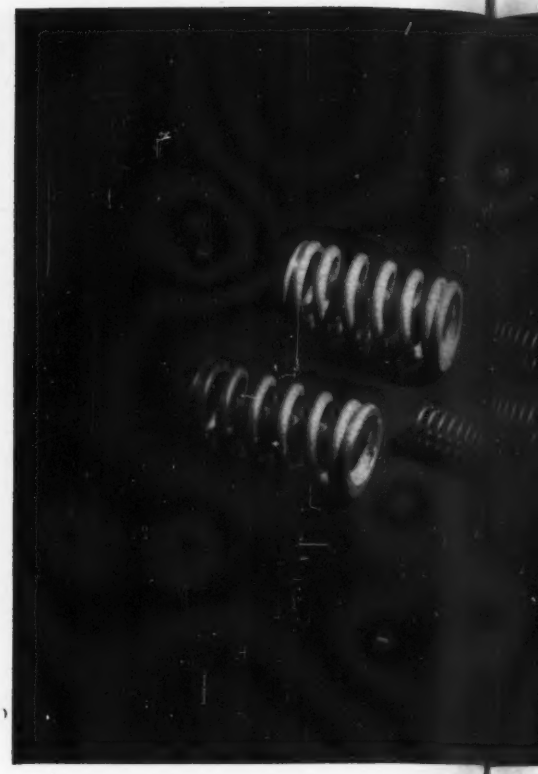
When the stop button is pressed, or if the protecting switch functions because the shuttle has not arrived in the opposite shuttle box at a precise instant, contactor *B* opens. This impresses a direct-current voltage across two of the motor leads through the full-wave rectox rectifier and braking torque, as shown in *Fig.* 1, is developed. The braking torque can be varied by adjusting the resistors *R*1 and *R*2. With this scheme the loom is easily brought to rest in ample time for protection stop, smoothly and without shock. Contactors *D* and *M* open after the loom has stopped.

Electronic relays are being used for the safe, fast and positive stopping of machines when single threads break, or where bobbins in loom shuttles run out, and the like. *Fig.* 5 shows a single unit type of electronic relay. The drop wires are small open metallic contacts which, when held up by individual threads, keep the grid of the thyratron negative so the tube cannot fire. As soon as a thread breaks and its drop wire falls, the grid becomes positive and the tube fires, energizing the relay. This action is dependable even though the drop wire contact is made only for a few microseconds and the contact resistance, due to dirt and corrosion, may be as high as 30,000 ohms. The voltage across the open drop wire is about 10 volts with one side grounded, so open contacts are not dangerous. At the time the drop wire contact is made the current is so small that there is no arcing or burning on the lightest of drop-wire or contact-making schemes.

A similar electronic relay, but of the multiple-circuit type, is shown in *Fig.* 6. As many groups of drop wires as desired are connected as shown, one group to *D*1, one to *D*2, and so on, using as many thyratrons as there are drop-wire, or indicating groups. If a contact is made at *D*1 its thyratron will fire and pick up the relay and also illuminate the indicating lamp 1. Since the thyratrons are operated by direct current, once they are fired they will remain conductive and keep the relay picked up and the lamp illuminated even though its drop-wire circuit is opened. The relay is dropped out by a reset circuit in series with the relay coil, not shown in *Fig.* 6.

When any one of the drop wire circuits, such as *D*1, has fired its thyratron, current will flow through resistor *R*3 and because of the voltage drop in *R*3, the cathodes of all the thyratrons will become more negative than the grounded lead of all the drop-wire circuits. Therefore, after the relay has picked up and stopped the machine, should any or all of the remaining drop wires close they cannot cause their respective thyratrons to fire, and only the lamp corresponding to the drop wire that first made contact will be lighted. Hence the operator will know exactly where the trouble originated.

Selecting Protective Finishes for Springs



By Ronald F. Pond
Chief Spring Engineer
Reliable Spring & Wire Forms Co.
Cleveland

WHEN the designer approaches the question of determining the finish to be applied to a new spring, his choice may be complicated by as many as eight factors. That is, in dealing with some types of springs, as many as eight phases of the problem should be considered before the correct decision can be made, particularly when the objectives of desired appearance, corrosion resistance, and maximum economy are to be attained.

Springs are more difficult to finish than almost any other class of product. Infinitely varied in shape and proportion, all too many springs are poorly adapted to economical methods of handling in the finishing shop, such as barrel plating in the electroplating shop, and centrifuge application of finish in the paint and Japan shop.

Some springs, like those in *Fig.* 1, can be barrel plated at moderate cost if they do not tangle excessively. Another difficulty often encountered in plating, particularly with flat springs, is that heat-treating scale forms on the surface of the parts, which cannot be removed by commercial solvents.

Each type and kind of finish can be considered in the light of the results achieved by it, and of the difficulties of its application to specific groups of springs. Springs can be considered as groups, and the suitability of and difficulties in applying the various finishes can be related to the group. Regardless of the method of approach, a certain amount of repetition may occur, but this will be minimized if the designer will first consider the individual spring on his drawing board from the standpoint of what is to be achieved by the finish, and then take up the physical characteristics of the part in the light of the advantages, limitations, and difficulties of application of the finishes he feels will provide the desired appearance.

APPEARANCE: Proceeding according to this plan, the designer will in most instances begin by ascertaining what colors and finishes will be used on the parts adjacent to the spring in the assembly. He should then base his initial decision on whether or not the spring should have the same, or a contrasting color. At this point, a cautionary note should be stressed: Supposing that the spring is clearly visible and that customer acceptance is therefore an important angle, the designer should realize that he is going to complicate the task of procurement if he decides on a perfect matching of finish. Especially in the case of color-Japans, he cannot be sure of matching colors, if the spring source applies the finish, unless he specifies the paint by source, name and number, and furnishes data as to cleaning methods, number of coats, and baking procedures, down to the last minute detail. Though matching colors are justifiable sometimes, the practice is likely to increase cost and lengthen delivery time, for there is a good chance that a color and kind of finish will be specified which the spring source does not carry in stock. A contrasting color usually will give less difficulty.

Matching Colors Is Difficult

The problem of matching colors exists also with electroplated springs; identical plating procedures will produce varying results on machined parts and springs, depending on the surface finish of each be-

Fig. 1—Above—Left—Compression springs having an index between three and seven can be barrel plated at moderate cost

Fig. 2—Above—Right—Plated wire forms. Rings are bright nickel-plated oil-tempered wire, requiring exacting heat treatment to reduce hydrogen embrittlement. The rectangular shapes are precision formed and rack plated with hard chrome to resist abrasion

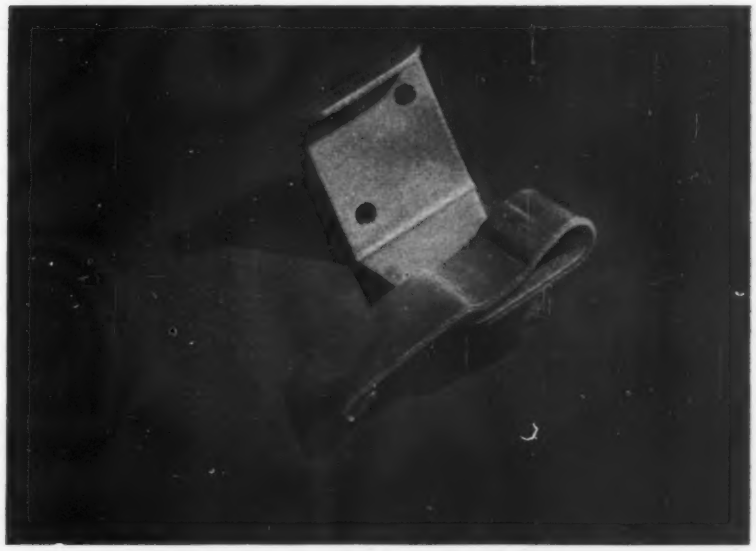


Fig. 3—Right—Folded shape cannot be cleaned in tumbling barrel. Electroplating also would not reach inside the folded section. Commercial solvent cleaning and phosphate type coating are recommended

fore the plating operation. In this connection it should also be mentioned that the appearance of certain electroplates differs considerably from the usual appearance of the base metals, and even from the appearance of the same metals applied by different means, such as hot-dipping, and the surfaces obtained on precoated wires. Electroplated tin, for instance, has a dull white appearance and unburnished electroplated copper is a dull red.

PROTECTION AGAINST CORROSION: There is no question but that the most common reason for applying finish to springs is the frequent need for protecting the part against corrosion. While the corrosion-resistant materials—stainless steels and the nonferrous groups—regularly provide the best answer where service conditions are really tough, the high cost of these materials and the lower level of their physical properties force the specification of carbon steel for most springs. Carbon steel, if not protected by oil or otherwise, is capable of turning brown with rust overnight in a humid atmosphere, and even after nothing more than the touch of a sweaty hand.

Electroplating, it must be emphasized, is not a perfect corrosion preventive, though it is often so termed. This statement can be applied fairly equally, to all kinds of plating. In most atmospheres, cadmium offers the best resistance. Next in order of corrosion resistance would come zinc; then the expensive and more lustrous copper-nickel-chrome combination commonly applied to automobile trim; copper; tin; nickel (by itself); and brass. The sequence,

it must be emphasized, will vary somewhat in different climates and applications.

The degree of corrosion resistance depends on the thickness of plate deposited and also on the porosity of the plate. Consideration also should be given as to whether or not the material surface is suitable for an even plate, that is, free from pits and oxidation or scale. The most common high-protective plating for springs is 0.0003-inch of cadmium, which is generally considered to be capable of withstanding 100 hours of salt spray. Maximum corrosion resistance is stated by most authorities as being at the level of 200 hours of salt spray; for this amount of protection, a plating deposit of 0.0005-inch of cadmium is most frequently specified. Recent developments have brought the protective level of Irridited or chromated zinc to approximately the same point. As to the copper-nickel-chrome, and nickel-chrome combinations, 0.0002-inch of copper plus nickel or 0.0001-inch of nickel plus 0.00001-inch of chrome will resist 96 hours of salt spray without appreciable corrosion.

As to corrosion resistance, the value of the other common plates, nickel (alone), copper, brass, tin, and silver drop away sharply, and these finishes are used largely for some of the other purposes discussed in this article.

Oil Gives Best Protection

At this point it may be well to point out that the best protective finish for carbon-steel springs is oil, as long as the part is used in a manner and location permitting it to be so protected. Of oil finishes there are many, ranging from SAE 10 petroleum, through various commercial products carrying waxes and having various drying and adhesion potentials. By specifying such finishes (it is well to specify alternatives to help cost and delivery), the designer is protecting his new spring against wear, and at the same time is specifying a corrosion inhibitor which will in no way harm the spring itself.

PROTECTION AGAINST ABRASION: Allied with the factor of protecting the spring itself against wear, and consequently the mating surfaces involved, is the need sometimes encountered in flat springs and wire forms for obtaining a surface which will resist abrasion and galling, *Fig.* 2. For such purposes, the ideal electroplates are decorative chrome (copper-nickel-chrome combination) and hard chrome, as well as tin, lead (which also is resistant to sulphuric acid), and copper.

In applications where a black or dull gray finish is satisfactory, the designer has available various well-developed commercial phosphate treatments, and oxide finishes such as Pentrate, all of which achieve a surface grain structure, retaining the oil applied in the course of treatment. All offer good corrosion resistance. Of these Parco-Lubrite (Parker Rust Proof Co.) is especially to be recommended. Twenty-four hours of salt-spray resistance can be expected.

KIND OF MATERIAL: The careful designer should next consider finish from the standpoint of the kind of material he has selected for the spring and be ready to reconsider that choice. Electroplating of oil-tempered spring wire is inherently difficult. This material, because its finish generally requires harsher cleaning agents prior to the plating bath and because of its granular structure, is susceptible to hydrogen embrittlement during the plating process. Frequently this cannot be removed by subsequent heat treatments and a great deal of spring breakage can be charged to this condition. The fact that oil-tempered wire springs are sometimes plated successfully, *Fig.* 2, should not be relied on as a safe guide in new decisions.

This principle often applies to flat springs which are most frequently made from annealed steel, hardened and drawn after all forming operations. The

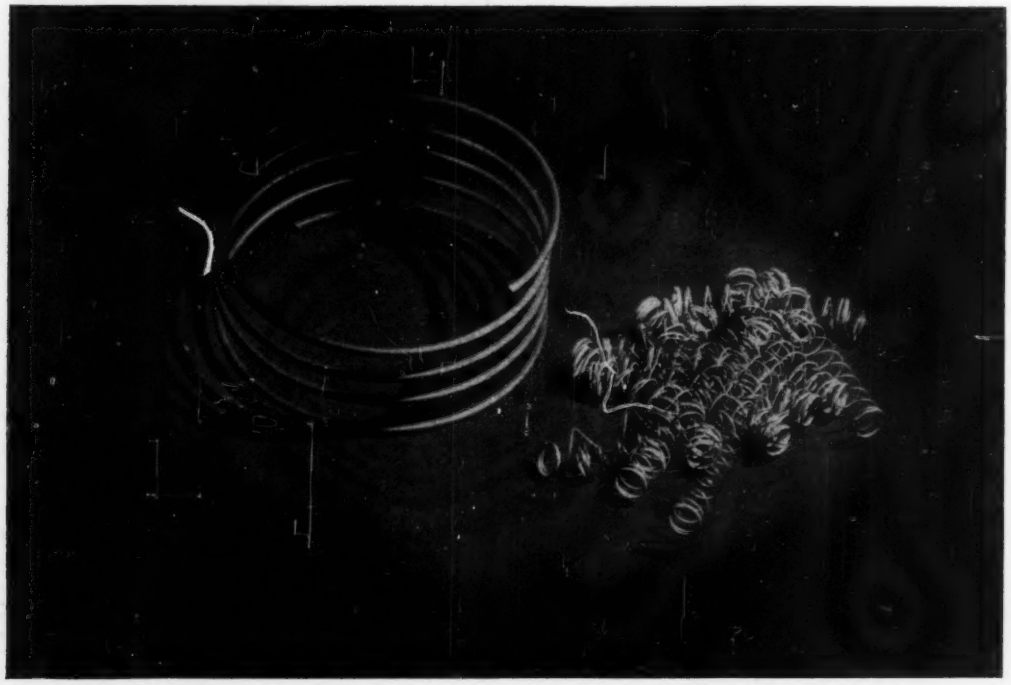


Fig. 4—Large spring index makes these springs practically unplatable in tumbling barrel. The mat of tangled springs of large index and small wire size cannot be unscrambled without damaging the product even though springs have closed end coils. Corrosion-resistant or preplated wire is recommended

heat treatment results in a grain structure similar to that of oil-tempered spring wire. Consequently, it accepts a considerable amount of hydrogen embrittlement, especially when plated with cadmium, zinc or the copper-nickel-chrome combination. In this type of work, however, embrittlement can be effectively controlled, *Fig.* 2, if the Rockwell hardness can be reduced to the lowest possible range consistent with the deflection requirements of the part and if drawing time can be increased, even at a lower temperature.

Because brass and phosphor bronze oxidize rather quickly, becoming dark and dull in finish, nickel or cadmium sometimes are applied to improve appearance. This practice usually has satisfactory results unless a plating of nickel heavier than 0.0002-inch is applied. Should this happen, the effect on the load rate of the spring may be surprising. Nickel, being a hard and brittle plate, will stiffen a spring appreciably and thus upset the best calculations of the designer.

SURFACE OF MATERIAL: Even more important than the kind of material, when electroplating is required, is its finish. The ultimate in a lustrous plate cannot be obtained on a rough and pitted surface. If a flat spring is applied where finish will have an effect on consumer acceptance, the designer should specify the best commercially obtainable finish for the material in question.

Springs and wire forms made from the lower priced basic, hard-drawn and oil-tempered carbon-steel spring wires will carry less luster when plated than will the more highly polished music spring wire or drill rod. Therefore the latter materials should be specified in applications requiring the brightest possible finish.

Depending on the design of the parts, however, the cheaper materials may sometimes be given treatment after forming which will materially improve the brightness of the plate. Parts which will not tangle and which are sufficiently sturdy in section to withstand rough treatment without distortion may be tumbled in sawdust or fine abrasive, with a much brighter plate as the result. Sand blasting before plating is often necessary to remove heat-treating scale from flat springs, particularly when the part is so formed that its inner surfaces cannot be reached effectively by the abrasive in a tumbling barrel, *Fig.* 3. This treatment is capable of removing scale completely and makes possible an even plate with perfect adherence. The resulting finish, in the case of both zinc and cadmium plating, is satin rather than highly lustrous. When nickel or copper-nickel-chrome is applied to such a surface, the finish is bright. On close inspection, however, the surface is seen to be slightly uneven and finely pitted, due to marking by the sharp particles of sand.

Heavy Plating May Crack

DEFLECTION REQUIRED: Particularly in the case of nickel and chrome plating, the deflection required of the spring must be considered. These plates, being hard and brittle, often will not withstand torsional twist and bending which may be well within the potential limits of the material used. In order to obtain high luster, nickel and copper-nickel-chrome thicknesses of 0.0002 to 0.00075-inch are often specified. If the heavy plate cracks and flakes, the spring source and plater should not be blamed on the grounds of inferior workmanship.

SIZE OF SPRING: The designer should bear in mind that barrel plating is more economical by far than still-tank plating. The difference in cost, particularly for small springs, is so great as to make the cost of still-tank plating prohibitive. As the size of the spring increases, the difference between barrel and still-tank plating costs narrows until the point is reached where a spring is too large to be handled in a plating barrel efficiently. This breaking point is difficult to determine exactly but it might be said

Fig. 5—Corrosion-resistant wire should be utilized for these springs because of difficulty in plating. Some are small enough to catch in holes in plater's barrel, others have large index. If plated, closing the open hooks somewhat would reduce trouble

that, where the overall length of a spring exceeds ten inches, still-tank plating may be the only or cheaper method.

PROPORTIONS OF SPRING: Spring engineers usually agree that a ratio of one to seven is an ideal spring proportion, *Fig.* 1. That is, in coiled springs made from round or square wire, the ideal proportioning of a spring would dictate that its mean diameter should be seven times the diameter or section of the wire. Applied to the problem of electroplating, the rule can be revised to this extent: from the standpoint of plating and plating cost an index range of three to seven is ideal. Above this range, any helical spring becomes increasingly pliable and is subject to tangling and distortion when subjected to rolling and agitation in the plater's barrel. The index range of 13 to 24, for instance, *Fig.* 4, should be regarded definitely as a danger zone.

When the effect of tangling is distortion of the spring, the danger is obvious, but the story is not told by this statement alone. The spring manufacturer, necessarily mindful of all the costs incident to the production of springs, will figure his prices to include the higher labor costs necessary to handle, through the various phases of the manufacturing process, a spring which tangles easily. Sometimes he must charge for disentangling and inspecting the parts after plating.

Costs May Become Excessive

A helical spring with an index of three to seven may cost seven cents per pound to cadmium plate to a thickness of 0.0002-inch. A spring of the same weight, but with an index of twenty may cost more than twice as much to plate to the same thickness. Moreover, when springs tangle badly they become enmeshed into a dismaying mass of tangled wire, *Fig.* 4, in the center of which may be some incompletely plated springs, even some entirely devoid of plate.

When such conditions can occur, the idea of electroplating the part after forming should be abandoned. The spring should be redesigned to use a corrosion-resistant material, if corrosive attack is to be severe and space will permit use of lower tensile materials such as phosphor bronze and brass, *Fig.* 5. In such instances stainless steel is by far the best answer and may offer real economy when the price of the stainless steel spring is compared to its counterpart made from carbon steel, plus plating.

Preplated Wire Is Alternative

Another convenient alternative is to specify preplated carbon-steel wire, *Fig.* 6. At present cadmium, zinc (galvanized wire), copper, lead alloy, and tin (subject to government order) plated wire are available. These preplated materials have limitations of some importance. There is no protection at the butt end of the wire. The plating on the wire commercially available will not meet government plating specifications, as it seldom reaches a thickness as great as 0.0001-inch. The finish is subject to damage through scratches and tool marks during coiling and forming operations. Particularly with respect to tin, the luster and effectiveness of the plate are reduced by the heat treatment generally required for helical springs subsequent to the coiling and forming operations.

Decidedly worthy of comment is the fact that a prenickel-plated low-carbon basic steel wire, suitable for wire forms but not springs, is now available on which the plating is not brittle, has high luster and is extraordinarily resistant to fracture through tool marks and scratches, *Fig.* 6. A spring wire, having the same finish, is under development.

It is obivous that springs tangle for other reasons

(*Continued on Page* 194)

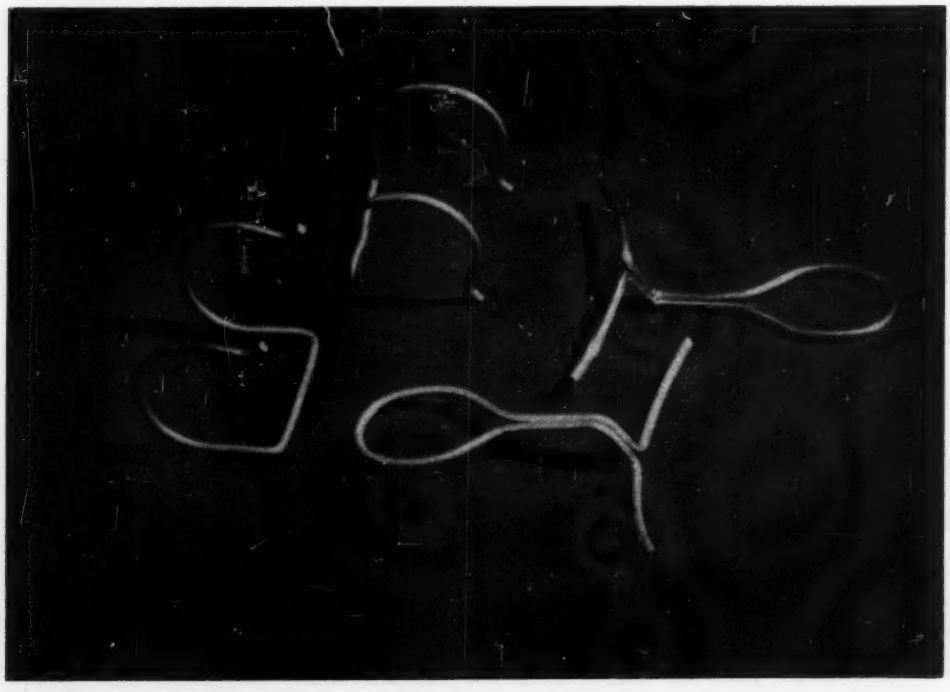
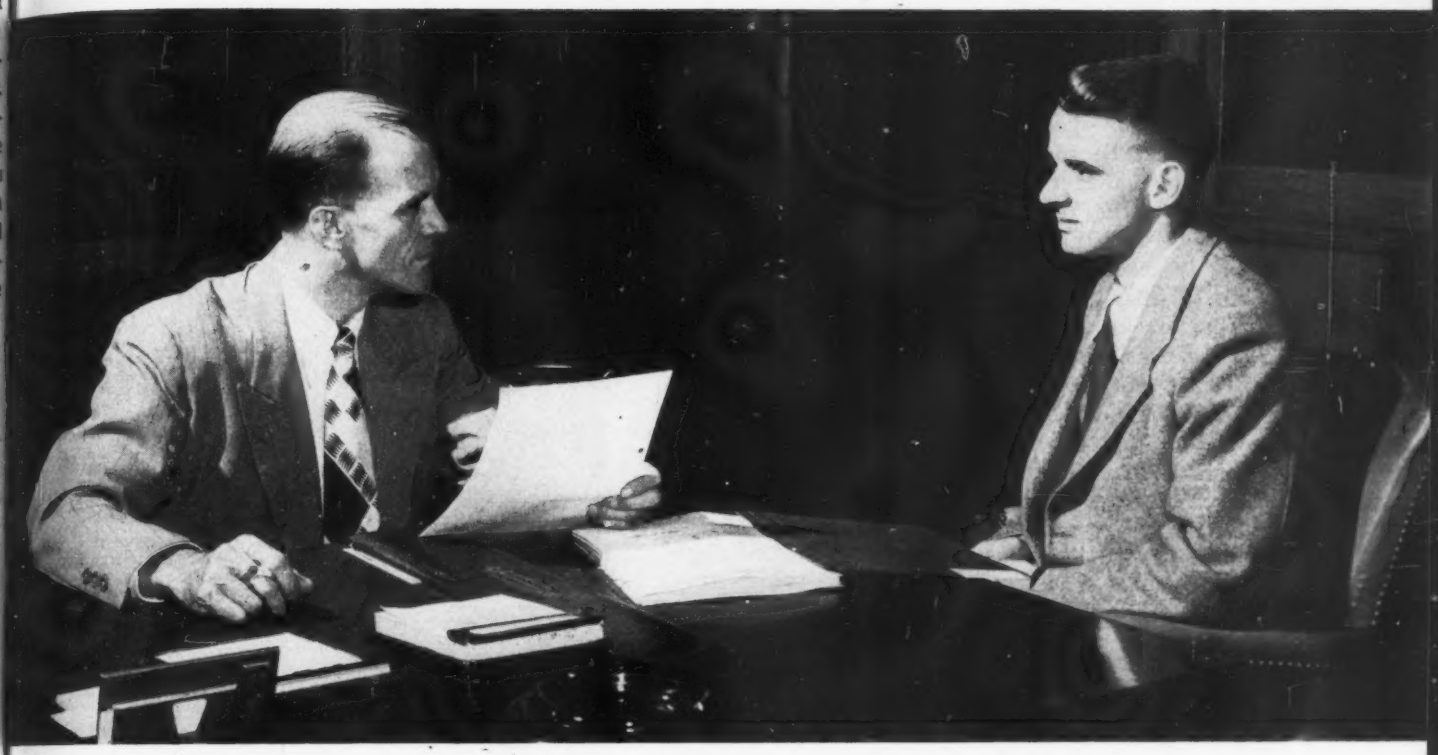


Fig. 6—Preplated wire is economically used for these forms because of difficulty in handling during plating



Selecting Engineering Personnel

Proper selection of personnel offers the key to maintaining the vitality of the up-to-the-minute engineering department

By T. W. Prior
Personnel Department
Goodyear Tire & Rubber Co.
Akron, Ohio

THE life-blood of the engineering department in any industrial organization, large or small, is the influx of capable new personnel, bringing along with them a flood of new ideas and new enthusiasm. One of the most important functions of any business, therefore, is competent, conscientious hiring of new engineers and designers.

Modern industry is becoming more and more specialized in all of its functions—engineering, sales, research, etc. Colleges and universities throughout the country, in order to meet this trend toward specialization, are offering special courses to provide the correct background for the industrial leaders of tomorrow. Thus, the selection of new personnel to fill specialized jobs within industry is in itself becoming more and more complicated.

Any company representative who is called upon to select new engineering personnel for jobs in industry must be more than a "jack-of-all-trades." He must be well versed in all the varied requirements of the engineering department or departments in order to be able to recognize good material. No longer may the personnel representative depend on a record of academic achievement alone—although we do not

Fig. 1—Whether experienced or inexperienced men are concerned, the first step is usually the personal interview

mean to minimize the importance of thorough scholastic training. The engineering graduate who hires into a large organization today must carry his weight not only in his specialized field; but if he is to go far, he must be something of a diplomat, a salesman and a manager as well as an engineer. Opportunities for the researcher to become lost in his laboratory with his test tubes, for instance, never having to meet the public or to deal with administrative problems or sales problems within his own organization are few and far between.

All of this places an additional burden on the personnel representative. He must hire not only people with good technical training, but people who are well balanced in almost every field. Since 1939 an additional problem has been added to the personnel representative's job in that the demand for compentent specially trained men has increased by leaps and bounds.

Success Still Depends Upon Individual

In this article, some of the pertinent aspects of personnel interviews and hiring for present-day industry will be discussed. It is important to point out, however, that the repetition of the old American success story where a man with little formal education can make the most of his opportunities and become another Edison or Rockefeller must not be precluded. In fact, the president of Goodyear is an excellent example of just such a success story. The important fact remains, however, that if the average business is to survive, it must have a good "pipe-line" from the sources of formal training—colleges and universities of the nation—to the work benches of practical knowledge within the particular organization.

This article, therefore, will attempt to give an insight on some of the problems facing the personnel representative and will endeavor to show at least one method of approach to some of them. The problems involved in the making of a well-balanced, well-trained new employee into a smooth functioning credit to the particular organization into which he hires —a matter of proper training and indoctrination—is the ammunition for another interesting article.

Personnel functions are usually considered to be divided into three sections: Selection, training, and personnel relations. In this article, discussion will be confined to problems and procedures in the field of selection and the contacting of prospects.

There are several commonly recognized sources for personnel which may offer possibilities; (1) People who apply in person (2) people who apply by mail, and (3) people who are recommended by employees and friends. In addition, the employer searching for qualified engineering personnel utilizes such channels as employment agencies, advertisements, other industries that have people working on the same or similar types of work where the experience would be valuable, and college placement services.

Contacting of colleges by industrial concerns has increased by leaps and bounds in the last twenty-five years. In the June issue of *Nation's Business* an

article by Homer H. Shannon, "The Bull Market i Mortar Boards" brings out the tremendous increas in college recruiting in the last year and shows th there are prospects for even more activity in thi field for the future.

Selection procedures, therefore, may stem fro a number of sources; and the people who are bein considered usually fall into one of two groups: Ex perienced or inexperienced. Selection interviews fo experienced and inexperienced prospects naturall are different.

Approach Depends Upon Applicant

In considering experienced prospects, the discus sion is inclined to center largely around work an on-the-job training; whereas, in considering the in experienced applicant, in most cases graduates i whom is sought fundamental engineering prepara tion and personal potential, the interview cente around academic training, job interest and perso ality.

In either group, selection usually begins with personal interview, *Fig.* 1. Many companies ha supplemented their personal interviews with examin tions and tests. In many fields, these have not be proved—especially in the industrial field where th are still considered pretty much experimental. So companies, however, have been very successful wi aptitude tests. Procter and Gamble, which has us such tests for a number of years, is probably th outstanding industrial concern to carry on work this field.

Considering of the mechanics of interviews, the are threee things sought for in an applicant, whet

Fig. 2—In interviewing college men, the first step is usually a group meeting at which the general aspects of the jobs being offered are given

inexperienced or experienced; What kind of an *individual* is the applicant? What does he *want* to do? What *can* he do?

First of all, for example, assume a hypothetical case involving an experienced prospect. Within the organization are openings for a number of technical men—machine and tool designers, perhaps. The first step is a look into the present organization to see if there are already available men who are prepared in the field, but who are doing some other or subordinate work. If this is true, selection of prospects is largely a matter of reviewing their experience and obtaining assurance that the men are worthy of advancement or transfer. But if acceptable prospects are not among present personnel, then the outside sources mentioned previously must be utilized.

The selection procedure is begun by asking the applicant to fill out a standard application form. This application alone could be the subject of a complete article in itself and so cannot be passed over without some discussion. Basically, the application must be simple, it must be short; and it must bring out the points of interest which will assist in providing a springboard for the personal interview. The purpose is to find out what the man looks like on paper.

The qualification blank used in considering the college graduate brings out his personal background, his educational background, and his interests. A section which should not be overlooked in this particular type of form is the extra-curricular activities. The college interviewer must be experienced in fraternities, honorary organizations, etc., so that he can spot the man's achievements without asking him a great many questions. In considering an applicant, his qualifications are judged in the following proportions: Forty per cent personal make-up, forty per cent training and scholastic record, and twenty per cent interest or desire.

A special interview guide is designed to be used by an interviewer, particularly an inexperienced interviewer, such as a department head, who does not spend most of his time interviewing people and who is not particularly trained in personnel techniques. This guide is helpful in learning good interviewing procedure. In this guide, in addition to the personal and educational qualifications, the financial and experience records are stressed and much more information is asked for than that which is given on the qualification blank. The aim is to determine why a man did certain things and whether or not these things fit into a logical pattern which may assist in evaluating the applicant relative to the requirements.

In considering experienced personnel, a written summary of the man's past record is valuable. Many companies have a rule which is good. It provides that in addition to writing reference letters, a check of one or two of the previous employers be made by telegram or telephone. Almost without exception it is found that previous employers will give a true story and a realistic picture of the man.

In considering the second or inexperienced group, selection of young engineers is concentrated pretty largely in the field of college recruiting. Most companies, in sending out personnel representatives to colleges have found it necessary to send men who are well versed in the requirements for almost all of the company's operations. This is true because in addition to selecting new employees, the personnnel representative must do a certain amount of public relations and educational work for his company.

Interviewing college men is begun with a group meeting, *Fig.* 2, in which an explanation of the jobs being offered and the training to be given is made. Perhaps some company literature is passed out.

Interviews Require Skill

Individual interviews begin by offering the applicant information concerning the job rather than going directly into the applicant's own personal qualifications. Then, after this brief discussion, an attempt is made to determine the qualifications of each applicant individually. What kind of an individual is he, and what can he do? This type of interview requires considerable experience and skill on behalf of the company representative. The primary objective is to find potential engineering personnel for the organization and little or no information on similar experiences is available for guidance. The professors who have associated with the applicant for a period of four or five years usually can help considerably.

What does the man want to do? This is almost always answered by the applicant's reaction to the statement: "I have an opening for a design engineer. Do you feel this type of work is what you are looking for?" If the applicant is interested, his re-

actions will be not unlike an ardent basefall fan talking about his favorite game.

Some of the engineering schools, particularly sponsor academic-training-actual-job-experience cooperative programs simply because they help the student find what he wants to do. Summer experience and family influence also have a direct bearing on what the applicant wants to do.

Success of most college interviews depends largely on the skill and planning of the interviewer. Usually, these interviews are short—fifteen to twenty minutes—and in addition to telling the company story, they help find the answer to the three questions: What kind of an individual is he? What does he want to do? What can he do?

Up to now in the selection of technical personnel, however, too much emphasis has been placed upon technical ability as determined by the grades the applicant received in school. Not enough stress has been placed on the man's personal make-up. Very few companies have any place where the individual worker is paramount. Men have to work pretty largely in teams, and we are as much concerned today with how well the man will function as a part of one of these teams, regardless of his background, as in his technical skill. As a result of past experience along this line, it is believed that the man who has the responsibility of selecting technical personnel will attempt to go deeper and find that the man is not only properly prepared but also will discover how well he is going to fit into the organization.

It would be well to mention four books which can be of help not only to the man who has the responsibility of selecting personnel but to applicants who are looking for jobs as well. Books such as Paul Boynton's *Six Ways to Get a Job,* Lyons and Martin's *The Strategy of Job Finding, The Choice of an Occupation* by the Yale Press, and *The Young Man in Business* by Howard Davis should be of interest to both groups.

Planetary Design Short Cuts Eliminate Tedious Analysis

To the Editor:

The author of the article "Modified Approach Permits Flexible Planetary Design" (M.D. Sept., 1948) is wrong in suggesting a standard planet pinion to mesh with a so-called Buckingham (short addendum) ring gear. The suggested combination will produce a poor gearset with excessive backlash, shallow working depth and low tooth contact ratio.

From Buckingham's *Manual of Gear Design,* Section II, Pages 108 and 111, the ring gear dimensions are: Root radius = 3.149 in.; internal radius = 2.940 in.; and arc tooth thickness = 0.1389-in.

Dimensions of a standard 22-tooth pinion are: Outside radius = 1.200 in.; root radius = 0.975 in.; and arc tooth thickness = 0.1571-in.

Backlash is equal to the circular pitch minus the sum of the arc tooth thicknesses: = 0.3142 − (0.1389 + 0.1571) = 0.0182-in. This is excessive since 0.003 to 0.005-in. is considerd normal backlash for ordinary applications.

Working depth is equal to the center distance plus the pinion outside radius minus the ring gear internal radius: = 1.900 + 1.200 − 2.940 = 0.160-in. This is only eighty per cent of full depth and reduces the tooth contact ratio from 1.600 to 1.439.

If a standard planet pinion is used, the ring gear should also be standard. The inside diameter, however, may be increased to 5.830 in. in accordance with Buckingham's recommendation. The dimensions of this ring gear are: Root radius = 3.125 in.; internal radius = 2.915 in.; arc tooth thickness = 0.1571-in.

In this combination the theoretical backlash is zero but the desired backlash can be obtained by cutting the teeth slightly deeper. The tooth contact ratio will be 1.754 instead of 1.439. If a tooth contact ratio of 1.600 is acceptable the inside diameter may be increased to 5.854 in.

It is stated under *Fig.* 2 that "standard pinion being smaller than Buckingham pinion provides more clearance." Actually, the clearance may be the same for either standard or special gears because the inside diameter of the short addendum gear is increased by the same amount as the outside diameter of the long addendum pinion, and the whole depth of the teeth is the same in both cases. The pinion arc tooth thickness is larger than standard by an amount equal to outside diameter increase multiplied by the tangent of the pressure angle. The gear arc tooth thickness is reduced by the same amount.

CARL A. JOHNSON
Springfield, Mass.

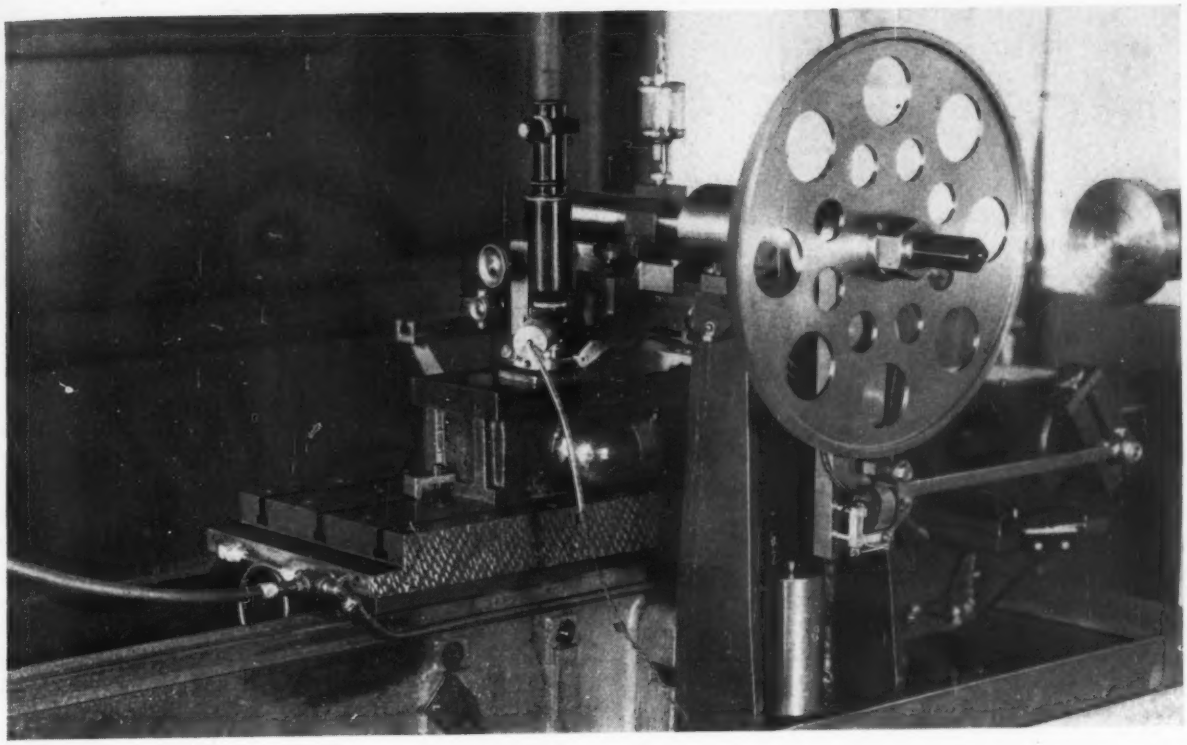
To the Editor:

The author realizes that the gear set recommended in his article is not the best, but it permits selection without going through a tedious analysis of interference calculations as outlined on Pages 36 to 40, Section II, of Buckingham's manual. This was the whole point of the article.

The gear set suggested by him would be better from the strength point of view but automatically calls for interference calculations which Buckingham modifications were designed to avoid, as the 0.025-in. difference in outside radii of the two pinions is removed by using a standard ring gear. Furthermore, a contact ratio of unity is considered good, inasmuch as larger values are theoretical and may not be realized because of flexibility of the supports of the gears, making it impossible to count upon a particular value.

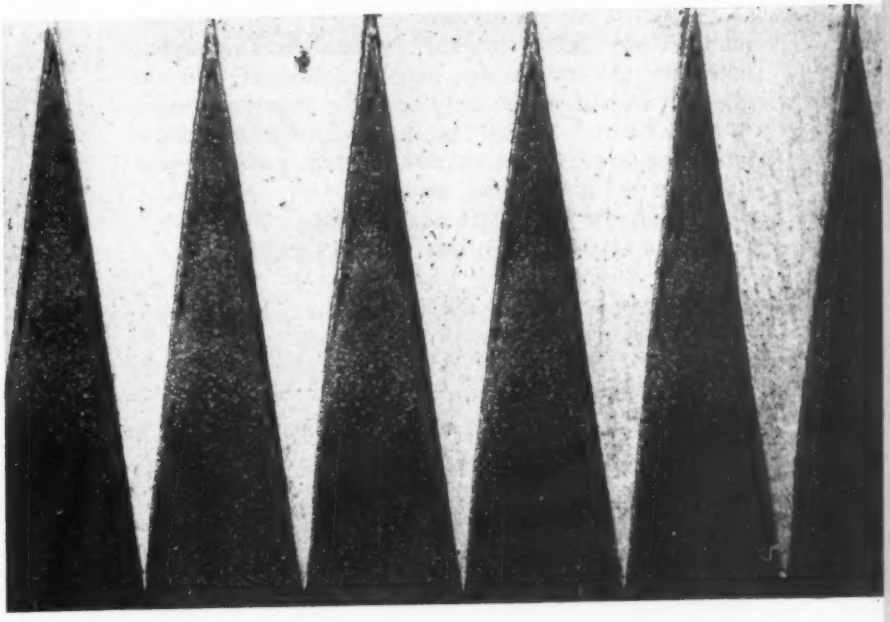
When strength requirements and wear characteristics are important, a careful selection is recommended and short cuts in the article should be avoided.

SERGEI G. GUINS
The Chesapeake & Ohio Railway Co.



Designating Surface Roughness

. . . with the aid of geometric standards



By Clayton R. Lewis
Staff Engineer
Chrysler Corp.

and Arthur F. Underwood
Head of Mechanical Engineering Dept.
General Motors Research Laboratories

DEVELOPMENT of surface finish specifications, embodying a high degree of precision has been a project on which engineers of General Motors Corp. and Chrysler Corp. have been working for the past two years. Basically, the problem revolved around the production of a master set of

Fig. 1 — Top — Fine-line ruling machine, developed by General Motors Research Laboratories, capable of scribing up to 10,000 parallel lines per inch on a gold master specimen

Fig. 2—Above—Taper section of geometrically ruled specimen having 160-microinch average peak-to-valley height. Horizontal magnification 400, vertical 10,000 diameters

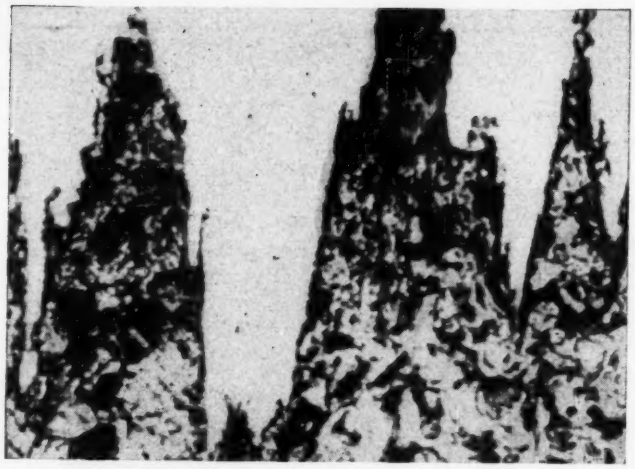


Fig. 3—Taper section of ground specimen, 148-microinch average peak-to-valley height. Horizontal magnification 300, vertical 7500 diameters

standard surface finish specimens from which exact replicas could be made. Inherent inconsistencies of ground, machined, lapped, or honed surfaces made it necessary to turn to other means of producing accurate master specimens. The almost infinite variety of surfaces makes it obviously impossible to reproduce all characteristic finishes in a limited number of sample blocks. In the final analysis, however, the roughness reading obtained from a given operation is the only criterion for acceptance. It was finally decided to make master specimens by ruling a geometric pattern on flat, polished gold plates with a diamond stylus.

No available equipment could handle such a microscopic ruling assignment, so it was necessary to develop a fine-line ruling machine which would rule grooves in a polished specimen up to 10,000 lines per inch while holding the pitch accurate to plus or minus 2 per cent. General Motors Research Laboratories perfected such a machine, *Fig.* 1, which is now in successful operation. The specimen is mounted on a table which reciprocates back and forth under a diamond stylus. This stylus is held in an arm actuated by means of cams which lower it onto the specimen with controlled pressure for every cutting stroke. Depth of scratch is controlled by adding or removing small weights to vary the pressure of the tool on the surface.

Indexing of the tool arm to scribe succeeding lines is accomplished with a carefully machined lead screw having 20 threads per inch, working in conjunction with a ratchet gear, precision cut with 500 teeth. This permits varying the density of the scribed lines between 250 to 10,000 per inch. The table is floated on oil to give smooth cutting and the entire machine is suitably mounted to minimize vibration.

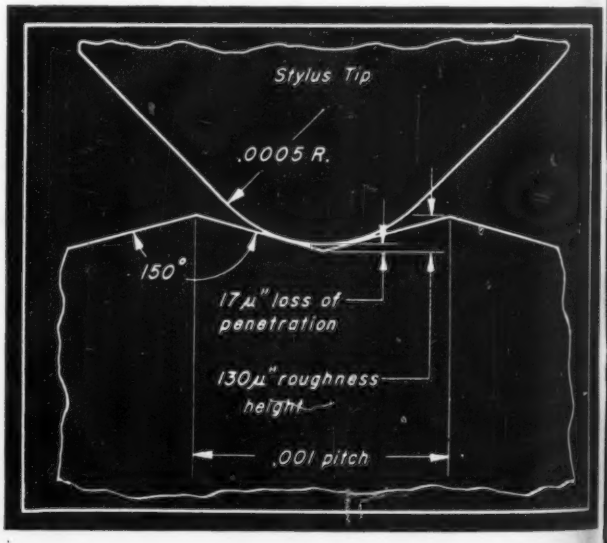
Ruled master samples are calibrated by the engineering staff of Chrysler Corp. This involves taper sectioning the scribed specimen across the lay of the rulings to amplify their contour by 25 times for easier measurement, *Fig.* 2. In *Fig.* 3 is shown a comparative specimen of a ground surface. This method follows a technique developed by H. R. Nelson of Battelle Memorial Institute, and is reported to be accurate to a fraction of a per cent on coarse rulings and to approximately 5 per cent on extremely fine rulings. Work done to date indicates a variation or error between individual rulings averaging 3 per cent.

TABLE I

Proposed Surface-Finish Standard

Depth of Cut (μ in.)	Angle of Cut (degrees)	Density (lines/in.)	Pitch (μ in.)	Loss of Penetration Due to Stylus Radius of (microinch) 500 (μ in.)	400 (μ in.)	100 (μ in.)
1000	126°54′	250	4000	59	47	12
800	136°24′	250	4000	38	31	8
600	145°36′	250	4000	22	18	4
500	150°	263	3800	17	14	3
400	150°	333	3000	17	14	3
320	150°	416	2400	17	14	3
250	150°	526	1900	17	14	3
200	150°	666	1500	17	14	3
160	150°	833	1200	17	14	3
130	150°	1000	1000	17	14	3
100	150°	1250	800	17	14	3
80	150°	1666	600	17	14	3
60	150°	2000	500	17	14	3
50	150°	2500	400	17	14	3
40	150°	3333	300	17	14	3
32	150°	3333	300	17	14	3
25	150°	5000	200	15	12	3
20	157°24′	5000	200	10	8	2
16	161°50′	5000	200	6	5	1
13	165°12′	5000	200	4	3	1
10	168°54′	5000	200	2	2	nil

Fig. 4—Diagram of stylus penetration in 150-degree geometrically ruled surface, vertical section



Replicas of the master samples are produced by electroforming, carried out by U. S. Rubber Co. The original scribed gold master is plated by the Ecco process about 0.001 to 0.002-inch with nickel and backed with 3/16-inch electrolytic iron. The plating is then stripped from the master specimen by mechanical means. Succeeding generations of replicas are made similarly, each from the immediately preceding specimen. Finally, the nickel surface of the replica to be used is given a flash plating of rhodium. As calibration is to be done on the sixth generation of replicas, considerable plating time naturally has

been involved. After the masters were calibrated, their first reproductions were mounted in a jig fixture and production plating began on the next generation of replicas. This phase of the work requires several months at best.

While the primary purpose of the work is to supply Chrysler and General Motors shops and engineering departments with an accurate set of surface-finish specimens, as soon as their requirements have been met, the specimen sets will be placed on the market at a cost involving only the expense of producing the electroformed replicas, and not any of the research cost underwritten by the two companies.

Specimen sets will be housed in a thin pocket-size metal case and will be removable from the case in groups of three or four roughness specimens to faciliate instrument comparison with production surfaces. Individual patch or plate size is ¾ x 1½ inches. Twenty-seven roughness values are provided in graduated increments from one-microinch (one-millionth inch, abbreviated μ in.) to 1000 microinches. These values are the average peak-to-valley distance measured as indicated in the diagram, *Fig.* 4, which shows a greatly magnified representation of stylus penetration in a ruled surface with 150-degree included angle and 0.001-inch pitch. Radius of the stylus tip is 500 microinches, the peak-to-valley roughness measurement is 130 microinches, with 17 microinches loss of penetration.

Data in TABLE I list various interrelated factors covering the range of the proposed surface finish standard, from 10 to 1000-microinch depth of cut.

TABLE II

Standards for Roughness and Waviness

Roughness (microinches)						
1	5	13	32	80	200	500
2	6	16	40	100	250	600
3	8	20	50	130	320	800
4	10	25	60	160	400	1000

Waviness (inches)					
0.00002	0.00008	0.0003	0.001	0.005	0.015
0.00003	0.0001	0.0005	0.002	0.008	0.020
0.00005	0.0002	0.0008	0.003	0.010	0.025

Roughness values less than 10 microinches ultimately will be calibrated. Pitch or peak-to-peak spacing of the rulings has been held between the limits of 200 to 4000 microinches to hold the standard surfaces within the frequency range of the Brush Analyzer and the Abbott Profilometer. The included angle has been made as wide as feasible to minimize error introduced by lack of penetration of the diamond stylus. TABLE I shows these losses for the various depths of cut.

Surface finish standard, as incorporated in General Motors' engineering standards manual, defines the purpose and scope as, "establishing definite classifications for various degrees of roughness and waviness, and for various types of lay in the geometrical irregularities of surfaces of solid materials." A set of symbols is provided for use on drawings, and in specifications and reports. Ranges for roughness and waviness are divided into a number of steps and the general types of lay are established by type characteristics. The standard does not define the degree of surface roughness and waviness or the type of lay most suitable for a specific purpose; neither does it specify the means by which any de-

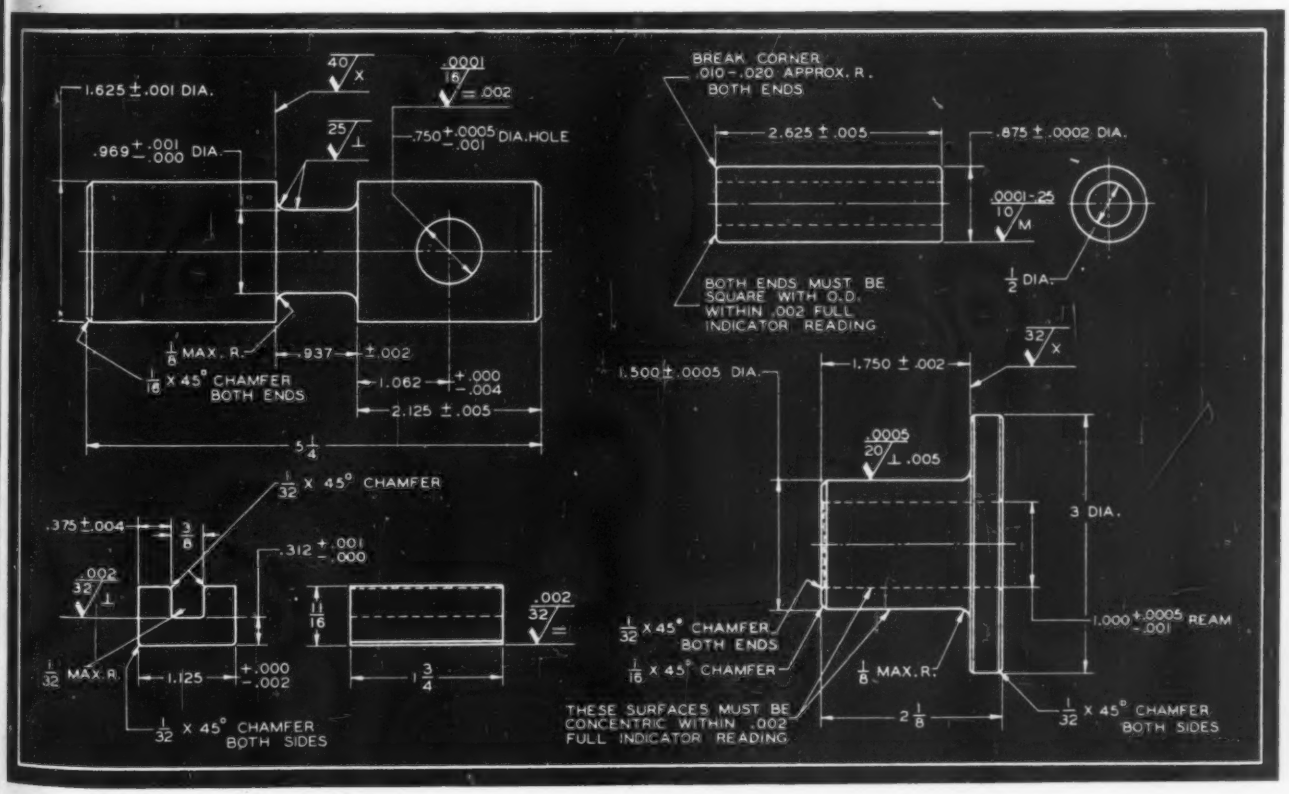


Fig. 5—Sample parts drawings showing typical applications of symbols and of surface finish specifications

gree of such irregularities may be obtained or produced. Furthermore, it is not concerned with the other surface qualities such as luster, appearance, color, corrosion resistance, hardness, microstructure, absorption characteristics or the like. The General Motors surface finish standard is written to conform to the American Standards Association B 46.1 standard. Definitions used in the standard for roughness, waviness and lay are as follows:

Roughness height rating: A physical measurement in microinches which shall be the arithmetical average of a series of peak-to-valley measurements of the surface roughness. These are to be taken with an approved roughness measuring method which has been calibrated with reference to the geometric surface finish blocks, and are to be taken normal to the surface and, unless otherwise specified, in the direction giving the greater value (usually across the lay).

Roughness width rating: The maximum permissible width of repetitive units of the dominant surface pattern. It may be specified in inches adjacent to the lay symbol. Irregularities having widths up to and including the maximum specified, or when no dimension is specified, up to and including the width of the irregularities due to machine feed shall be the basis for the roughness width specification.

Waviness height rating: A physical measurement in inches which represents the average vertical distance from peak to valley of the wave.

Waviness width rating: Waviness widths may be specified directly in inches.

Lay specification: The lay of a surface shall be specified by the lay symbol indicating direction of the visible surface marks.

Use of only one number shall indicate the maximum value of either the height or the width of irregularities. Any lesser degree shall be satisfactory. When two numbers are used they shall specify the maximum and minimum permissible value. Standards have been established only for height values as shown in TABLE II. Symbols for lay and for finish are identified in TABLES III and IV, respectively.

The symbol used to designate surface irregularities is the check mark, with extension, as shown in TABLE IV. The point of the symbol may be on the line indicating the surface, on the witness line, or on an arrow pointing to the surface. The long leg and extension shall preferably be to the right, as the drawing is read. The short leg may be increased in width to be more conspicuous if desired.

A method of comparison is employed for determining surface roughness. The piece being inspected may be compared by feel, or by instrument with a sample of known surface roughness. This sample may be a geometric surface finish block or a control piece part having surface characteristics in accordance with the specifications. Visual inspection is not reliable due to the fact that the piece being inspected may be of a different material than the sample and may have been machined differently.

When one piece has been machined and accepted in accordance with the drawing specifications, this piece may become the control piece, and all subsequent pieces may be compared with it visually and by feel. There will be occasions when the decision is marginal and it will be difficult to agree as to whether or not the piece conforms with the sample. This will be particularly true of the smoother finishes. In these cases, it will be necessary to use a profile measuring instrument.

Any standard device for linear measurement, bluing gages or light gages may be employed for surface waviness inspection. Measurement of waviness less than 0.0001-inch will ordinarily require the use of optical flats, bluing gages or similar devices. Inspection for lay is by visual method.

Illustrated in *Fig.* 5 are drawings of parts for which roughness of critical surfaces is specified. Standard symbols and designations for roughness height, roughness width, waviness height, and lay are used to exemplify proper practice.

TABLE III

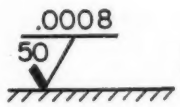
Standard Symbols for Lay

=	Parallel to the boundary line of the surface indicated by the symbol. Example: Parallel shaping, end view of turn and O.D. grind.
⊥	Perpendicular to the boundary line of the surface indicated by the symbol. Example: End view of shaping, longitudinal view of turn and O.D. grind.
X	Angular in both directions to the boundary line of the surface indicated by the symbol. Example: Side wheel grind, and traversed end mill.
M	Multidirectional. Example: Lap, superfinish.
C	Approximately circular relative to the center of the surface indicated by the symbol. Example: Facing.
R	Approximately radial relative to the center of the surface indicated by the symbol.
=.005	Numerical value in inches of the width of spacing or roughness is added to the right of the directional indication in the lay symbol.

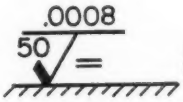
TABLE IV

Standard Designations for Surface Finish

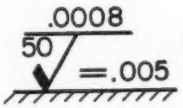
Roughness height value is placed adjacent to and on the inside of the long leg as shown.



Waviness height value, when required, is placed above the extension line as shown.



Lay designation, when required, is placed in the lay symbol under the extension line as shown.



Roughness width value, when required is placed to the right of the lay symbol as shown.

By W. Leopold
Engineering Dep't.
Wright Aeronautical Corp.
Wood-Ridge, N. J.

Rotating Disks at High Temperature

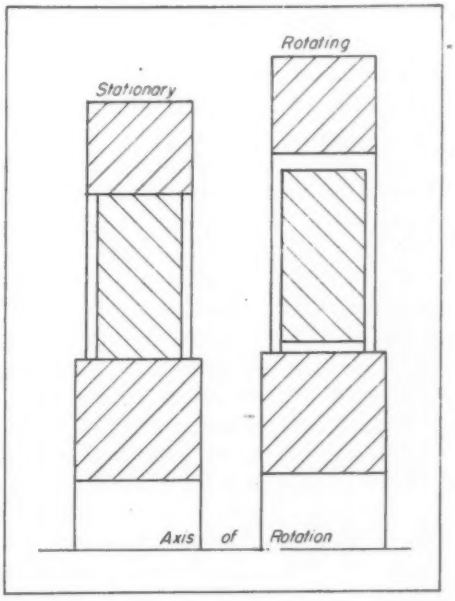
Fig. 1—Above—Turbosupercharger rotor for an aircraft engine. Temperature gradient and centrifugal stresses in rotor disk can be determined by method outlined in article

Presenting a simplified method of calculating centrifugal and thermal stresses

CALCULATION of stresses in a rotating disk is one of the most laborious problems of machine design. Analytical solutions which give precise answers are too lengthy for design practice. The sum-and-difference solutions in common use give accurate and rapid indications of centrifugal stress but, with the exception of certain rough approximations, neglect completely radial temperature gradients and resulting thermal stress. With the high temperatures used in most modern designs, particularly in steam and gas turbines, *Fig.* 1, the thermal stress and changes in the physical properties of the material with elevated temperature can no longer be neglected. The method presented in this article greatly simplifies the calculations and permits proper consideration of thermal stress.

When considering a disk under these conditions, it is sometimes confusing and disturbing to realize that the addition of material does not always strengthen the disk. To visualize this, assume any symmetrical rotating disk to comprise a series of concentric hollow cylinders. By allowing complete freedom of motion between adjacent cylinders, the effect of rotation may be illustrated by *Fig.* 2. It is known that the radial expansion and stress of a rotating cylinder is a function of the linear velocity of the outer radius. The effect, shown in *Fig.* 2, is that the expansion of the smallest cylinder is small in comparison to the expansion of the intermediate and largest cylinders.

Fig. 2—Below—Showing what would happen if rotating disk were made up of separate disks with freedom to expand



Remembering this, and restraining the disk to its actual condition of no expansion between cylinders, it becomes obvious that the hub will tend to support the entire rotating disk. From this, several conclusions may be drawn:

1. The hub is the restraining portion of a rotating disk
2. The larger the ratio of hub strength to disk inertia forces, the stronger the disk
3. A relatively strong hub restricts radial expansion to a great extent, thus reducing tangential stress.

These conclusions indicate that the hub is the place to put mass in a rotating disk to obtain greatest strength, when neglecting radial temperature gradients.

By similar reasoning, the effect of a radial temperature gradient may be seen. When a temperature gradient exists, the hotter material tends to grow radially and circumferentially, but is restrained by the cooler portions of the disk. Resulting compressive

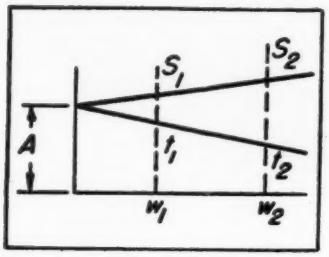


Fig. 3—Graphical method of representing Equation 5, which relates the radial and tangential stresses in a rotating disk

circumferential stresses in the hot material are superposed upon tensile circumferential stresses caused by rotation, thus determining the value of stress existing under the combination of the temperature gradient and rotation.

Value of the equivalent tensile stress attained by combining centrifugal and radial stress at any given point can be expressed as follows (see Nomenclature):

$$\sigma_o = \sqrt{\sigma_r^2 - \sigma_r \sigma_t + \sigma_t^2} \quad \ldots\ldots (1)$$

From this it is evident that large compressive forces resulting from a temperature gradient are detrimental to the strength of the disk since the term $-\sigma_r \sigma_t$ becomes positive, thus increasing rapidly the equivalent tensile stress. Because of this, and from examination of Equation 1, the most favorable conditions exist when the thermal compressive stress and centrifugal tensile stress combine to give a small positive tensile value. Since for a given temperature gradient the thermal compressive stress is constant, the disk must be allowed to expand until the resulting increased tensile stress equals the thermal stress. For the most common type of temperature gradient, decreasing from rim to hub, this would mean the rim would have a low value of stress while the stresses at the hub would increase slightly. A compromise must be made between the cold condition and that existing with a temperature gradient since, in order to allow the disk to expand radially as described, the hub would have to be weakened. The following conclusions therefore become evident:

1. A slight temperature gradient is beneficial as long as the slight compressive values obtained due to temperature do not become greater than the tensile stresses due to rotation
2. Large temperature gradients impose large values of compressive stress in the disk and can only be counteracted by allowing the disk to expand under centrifugal force
3. To increase the strength of a disk functioning under temperature conditions, directly opposite steps must be taken from those necessary to strengthen the disk without a temperature gradient
4. Low values of the modulus of elasticity and coefficient of expansion result in lower values of thermal stress.

In the following equations it is assumed that the temperature gradient along the radial length of the disk can be expressed as $T_x = T_i + kr_x^n$, where T_x is the temperature at any point x, T_i is the initial temperature at the absolute center of the disk, and r_x is the radius at point x. It is also assumed that all variables are symmetrical about the axis of rotation, while the disk is considered thin in that all axial stresses are neglected. The following expressions determine the radial and tangential stresses in a rotating disk of unit thickness without forces acting on the rim:

$$\sigma_r = \frac{(3+\mu)\ \gamma\ (\omega b)^2}{8g}\left[\frac{a^2}{b^2} + 1 - \frac{r^2}{b^2} - \frac{a^2}{r^2}\right] + \frac{E\alpha k(b^n - a^n)}{(n+2)\ (b^2-a^2)}\left[\frac{b^{n+2}-a^{n+2}}{(b^n-a^n)} - \frac{(b^2-a^2)r^n}{(b^n-a^n)} - \frac{a^2b^2}{r^2}\right] \quad \ldots\ldots (2a)$$

$$\sigma_t = \frac{(3+\mu)\ \gamma\ (\omega b)^2}{8g}\left[\frac{a^2}{b^2} + 1 - \frac{(1+3\mu)}{(3+\mu)}\frac{r^2}{b^2} + \frac{a^2}{r^2}\right] + \frac{E\alpha k(b^n - a^n)}{(n+2)(b^2-a^2)}\left[\frac{b^{n+2}-a^{n+2}}{b^n-a^n} - \frac{(n+1)\ (b^2-a^2)r^n}{b^n-a^n} + \frac{a^2b^2}{r^2}\right] \quad \ldots\ldots (2b)$$

Nomenclature

a = Radius of bore in disk, in.
b = Radius to rim of disk, in.
y = Thickness of disk at radius r, in.
ω = Angular speed of rotation, radians per sec
σ_r = Radial stress, psi
σ_t = Tangential stress, psi
σ_o = Equivalent simple tensile stress, psi
γ = Density of material, lb per cu in.
g = Acceleration of gravity, 386 in. per sec per sec
μ = Poisson's ratio
E = Modulus of elasticity, psi
α = Coefficient of linear expansion, in. per in. per deg F
n = Exponential power of r, expressing temperature gradient
k = Temperature proportionality constant
β' = Radial rotational constant
Y = Radial temperature factor
β = Tangential rotational factor
Z = Tangential temperature factor
$w = 1/r^2$

If there are uniformly distributed tensile or compressive forces acting on the edges of the disk, the stress due to them may be found using the theory of thick cylinders. When these are superposed on the stresses of Equation 2, the radial and tangential stress may be expressed as

$$\left.\begin{aligned} \sigma_r &= A + \frac{B}{r^2} - \beta'\omega^2 r^2 + Y \\ \sigma_t &= A - \frac{B}{r^2} - \beta\omega^2 r^2 + Z \end{aligned}\right\} \quad \ldots\ldots (3)$$

where

$$\beta' = \frac{\gamma}{g}\,\frac{3+\mu}{8}$$

$$\beta = \frac{\gamma}{g}\,\frac{1+3\mu}{8}$$

$$Y = \frac{E\alpha k(b^n - a^n)}{(n+2)\,(b^2-a^2)}\left[\frac{b^{n+2}-a^{n+2}}{b^n-a^n} - \frac{(b^2-a^2)r^n}{(b^n-a^n)} - \frac{a^2b^2}{r^2}\right]$$

$$Z = \frac{E\alpha k(b^n - a^n)}{(n+2)(b^2-a^2)}\left[\frac{b^{n+2}-a^{n+2}}{b^n-a^n} - \frac{(n+1)\,(b^2-a^2)\,r^n}{b^n-a^n} + \frac{a^2b^2}{r^2}\right]$$

A and B being constants of integration which may be calculated in each particular case. Using the notation

$$\left.\begin{aligned} s &= \sigma_r + \beta'\omega^2 r^2 - Y \\ t &= \sigma_t + \beta\omega^2 r^2 - Z \\ w &= \frac{1}{r^2} \end{aligned}\right\} \quad \ldots\ldots (4)$$

Equations 3 become

$$\left.\begin{aligned} s &= A + Bw \\ t &= A - Bw \end{aligned}\right\} \quad \ldots\ldots (5)$$

If s and t are known for any point of the disk, their magnitudes for any other point can be easily obtained by using the following graphical method developed by R. Grammel[1]. Let s_1 and t_1 denote the magnitudes of s and t for the point where $w = w_1$, as shown in *Fig.* 3. Then the magnitudes s_2 and t_2 for any other point where $w = w_2$ are obtained from the intersection of the vertical line through w_2 with the lines s_1 s_2 and t_1 t_2 which have their point of intersection on the vertical axis of the coordinates ($w = 0$) and are equally inclined to this axis. These lines represent Equations 5 graphically, having the common ordinate A on the axis $w = 0$ and having equal and opposite slopes $+B$.

To apply the foregoing to a disk of variable thickness, the actual profile must be replaced by a system of disks of uniform thickness. The stresses in the individual disks are calculated as previously described. Conditions where abrupt changes of thickness occur may be revised as follows. If y and $y + \Delta y$ denote the thickness of the two disks under consideration, the corresponding change $\Delta\sigma_r$ in the magnitude of the radial stress is described as

$$\Delta\,\sigma_t = \mu\Delta\,\sigma_r$$

From Equations 4

$$\Delta\,\sigma_r = -\frac{\Delta y}{y+\Delta y}\,\sigma_r$$

From the condition that the unit circumferential elongation is the same on both sides of the section, the change $\Delta\sigma_t$ in tangential stress is found as

$$\left.\begin{aligned} \Delta s = \Delta\sigma_r &= -\frac{\Delta y}{y+\Delta y}\,\sigma_r \\ \Delta t = \Delta\sigma_t &= \mu\Delta s \end{aligned}\right\} \quad \ldots\ldots (6)$$

From Equations 4, 5 and 6, and the graphical construction, which can be done by proportion also, the stresses in a disk of variable cross section can be determined.

In order to demonstrate the use of the method described in the foregoing, the stress pattern for the turbine rotor disk shown in *Fig.* 4 will be determined. The disk is divided into a series of constant thickness cylinders for analysis. Material is Inconel X, speed of the disk is 8000 rpm, and stress at the rim is 25,000 psi due to centrifugal force of blades imposed

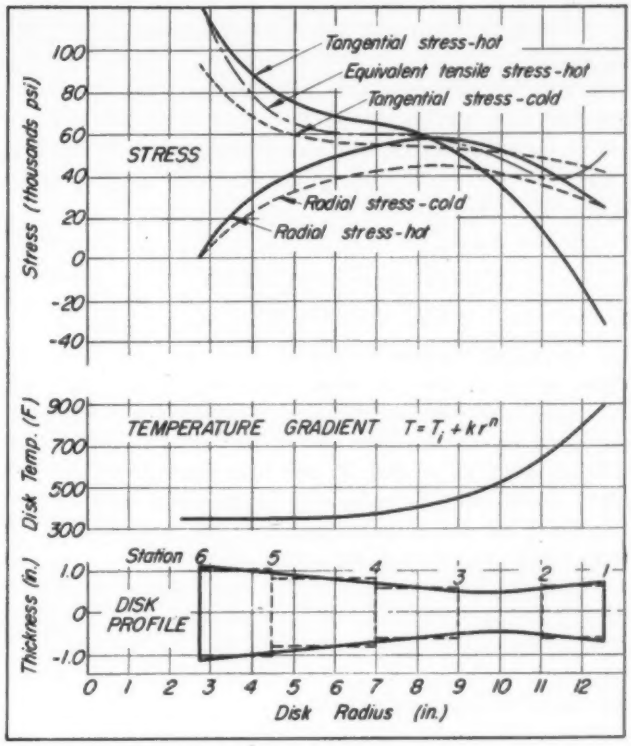


Fig. 4—Tangential, radial, and equivalent tensile stresses in rotating disk of profile shown, both cold and with temperature gradient indicated

[1] References are tabulated at end of article.

on rim surface. Temperatures are 900 F at the rim ($r = 12.5$ inches) and 350 F at the bore ($r = 2.75$ inches).

It will be assumed that the temperature gradient for this disk varies as the fifth power of the radius, ($n = 5$) although it is evident that any other value may be used as necessary to duplicate any desired temperature curve. Then

$$k = \frac{900-350}{12.5^5} = 0.001803$$

By assuming various values of r and using a constant value of k, it will be seen that the temperature at any radius r can be obtained, resulting in the temperature gradient curve shown in *Fig.* 4. Also, from these values of n and k, and the dimensions of the disk, the Y and Z factors of Equation 3 may be determined as variables in square and n powers of r and easily calculated for each station.

First ten columns of TABLE I are filled from these data, the stress calculation being begun at the wheel rim where σ_{r1} is given (Station 1). It will first be assumed, since σ_{t1} is unknown, that s and t are equal which gives, from Equation 4,

$$\sigma_{t1} = \sigma_{r1} + \beta'\omega^2r^2 - \beta\omega^2r^2 - Y + Z$$

or using values from TABLE I,

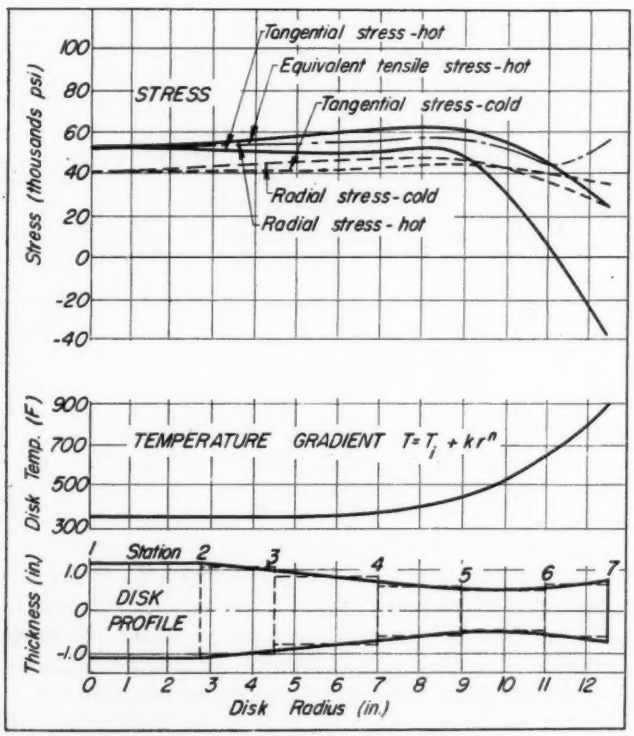
$$\sigma_t = 25{,}000 + 35{,}100 - 20{,}200 - 73{,}000 = -33{,}100$$

From Equations 4

$$s_1 = \sigma_{r1} + \beta'\omega^2r^2 - Y = 60{,}100$$

$$t_1 = \sigma_{t1} + \beta\omega^2r^2 - Z = 60{,}100$$

Fig. 5—Stresses in a disk without a central hole



Inasmuch as $s = t$, the s and t lines coincide in the construction of *Fig.* 3 and give the values of s_2 and t_2 at Station 2. Then from Equations 4

$$\sigma_{r2} = s_2 - \beta'\omega^2r^2 + Y = 40{,}110$$

$$\sigma_{t2} = t_2 - \beta\omega^2r^2 + Z = 11{,}400$$

To account for the abrupt change in thickness at this section, Equation 6 is used giving

$$\Delta s_2 = \left(-\frac{\Delta y}{y+\Delta y}\,\sigma_r\right) = 8022$$

$$\Delta t_2 = \mu\ \Delta s_2 = 2047$$

These values are added to the ordinates of the points s and t in *Fig.* 3, and by similar operations the values of s and t may be found for each section, thus completing the upper lines of columns 11 to 14 of TABLE I.

Since the tangential stress σ_t at the periphery of the disk was assumed arbitrarily, the conditions at the inner edge of the disk will usually not be satisfied. In order to satisfy these conditions, an additional calculation of columns 11 to 14 is required. It will be assumed that for this complementary solution, $\sigma_{r1} = 0$, $\omega = 0$ and temperature gradient equals zero. Taking an arbitrary value of σ_{t1}, a stress distribution is obtained, as before. Results of the calculation are shown in the lower lines of columns 11 to 14, TABLE I. The solution satisfying the actual conditions at the inner edge is obtained from combining the two foregoing stress distributions. It will be assumed that σ_{r6} and σ_{r6}' are the stresses at the inner edge of the disk, i.e., the hub which is Station 6, obtained from the first and second calculations respectively and σ_{r6}'' the actual stress at the inner edge. The term σ_{r6}'' is the value of stress at the bore obtained from a press fit of the disk on the shaft and is, of course, equal to zero for a loose fit. Solution for the actual condition of stress throughout the disk is obtained by superposing upon the first stress distribution (upper lines, columns 11 to 14) stresses of the second distributions multiplied by

$$c = \frac{\sigma_{rh}'' - \sigma_{rh}}{\sigma_{rh}''}$$

Average stress at the sections at which the thickness changes abruptly may be calculated as follows:

$$\sigma_r'' = \left(\sigma_r + \frac{\Delta s}{2}\right) + c\left(\sigma_r' + \frac{\Delta s'}{2}\right)$$

$$\sigma_t'' = \left(\sigma_t + \frac{\Delta t}{2}\right) + c\left(\sigma_t' + \frac{\Delta t'}{2}\right)$$

The results of such calculations for the case of zero radial stress at the inner edge are given in columns 15 and 16, TABLE I, and are plotted against disk radius in *Fig.* 4.

Values of stress for no radial temperature gradient may be obtained by merely recalculating the first

TABLE I
Tabular Solution for Disk with Central Hole

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Station	r	r^2	$10^3 w$	$\beta' w^2 r^2$	$\beta w^2 r^2$	Y	Z	y	$\frac{-\Delta y}{y+\Delta y}$	σ_r	Δs	Δt	σ_t	σ_t''	σ_r''
1	12.5	156	6.41	35100	20200	0	—73000	1.2	....	25000 0	.....	.....	—33100 2000	—32698	25000
2	11	121	8.26	27200	15700	7210	—32500	1.0	.200	40110 —290	8022 —58	2407 —17	11400 2290	13268	44056
3	9	81	12.35	18250	10500	12150	—194	1.2	—.167	63418 —997	—10600 166	—3180 50	50424 2923	49434	57932
4	7	49	20.4	11000	6350	13550	14350	1.6	—.250	61784 —2069	—15446 517	—4634 155	65614 4211	64171	53692
5	4.5	20.25	49.4	4550	2620	10650	23000	2.0	—.200	43384 —5753	—8677 1151	—2603 345	79864 8567	80343	37990
6	2.75	7.58	132	1705	983	0	38700		.....	3239 —15895	.....	.....	118306 20205	122436	0

trial, or upper line columns 11 through 14, and omitting the factors Y and Z. By combining with the second trial in TABLE I, and completing columns 15 and 16, the stresses for this case are rapidly obtained.

Values of simple equivalent tensile stress obtained due to the combination of the two principal stresses σ_r and σ_t may be obtained from Equation 1. The simple equivalent tensile stress is the criterion for yielding in materials of sufficient ductility where the maximum-shear theories are not considered applicable.

It will be noticed from *Fig.* 3 that stress is the ordinate and the reciprocal of the radius squared is the abscissa ($w = 1/r^2$). It is evident that for a solid disk, where at the absolute center the radius equals zero, the reciprocal of the radius squared approaches infinity. Recalling that in a solid disk the tangential and radial stresses are equal at the absolute center, and since the terms $\beta' \omega^2 r^2$ and $\beta \omega^2 r^2$ vanish at this point, while Y and Z become equal, it is obvious that for the hub, $w = \infty$ in *Fig.* 3, s and t must coincide and form a horizontal line from the $(N-1)$ to the N^{th} or infinity station on the graph. Therefore the method is reversed for a solid disk so that a stress of $\sigma_r = \sigma_t$ is assumed at the hub and, working from the hub to the rim, the stresses are calculated throughout the disk as in the previous problem.

To illustrate this, consider the disk shown in *Fig.* 5. This is the same as the disk shown in *Fig.* 4 except that it is solid. The calculations are begun by assuming $\sigma_r = \sigma_t$ at the absolute center and, in a manner similar to the previous example, obtaining a stress distribution throughout the disk. If the radial stress condition at the rim is not satisfied, a second trial must be made. If this fails to satisfy the rim condition, a third and final assumption may be made by assuming a straight-line proportion between the first two trials. TABLE II shows the completed tabular form sheet for this example.

The differences in *Figs.* 4 and 5 should be noted in the light of the statement made regarding temperature stresses at the beginning of the article. For the cold condition, it will be observed that the stronger hub of the solid disk reduces materially the stresses throughout the disk; on the other hand, under the influence of a temperature gradient, the stronger hub of the solid disk restricts radial expansion and thus induces a higher compressive stress at the rim.

Through use of the method described, a rapid and accurate analysis of any rotating disk is possible.

In the preparation of this article the author has drawn upon his recent ASME paper[2] from which *Figs.* 3 and 4 and TABLE I are taken.

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1. R. Grammel—"Einneues Versahren Zur Berechnung Rotierender Scheiben," *Dinglers Polytechnical Journal,* Vol. 338, 1923, Pages 217-219.
2. W. Leopold—"Centrifugal and Thermal Stresses in Rotating Disks," ASME Paper No. 48-SA-1.
3. S. Timoshenko—"Strength of Materials," D. Van Nostrand Co. Inc., New York, Vol. 2, 1941, Pages 236-258.
4. A. Nadai—"Theories of Strength," *Trans. ASME,* Vol. 55, 1933, Pages 111-129.

TABLE II
Tabular Solution for Solid Disk

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Station	r	r^2	$10^3 w$	$\beta' w^2 r^2$	$\beta w^2 r^2$	Y	Z	y	$\frac{-\Delta y}{y+\Delta y}$	σ_r	Δs	Δt	σ_t	σ_t''	σ_r'' !
1	0	0	∞	0	0	18400	18400	2.2	....	52040	.....	.....	52040	52040	52040
2	2.75	7.58	132	1705	983	18350	18300	2.0	.100	50285	5029	1509	51007	52799	51761
3	4.5	20.25	49.4	4550	2620	16200	15750	1.6	.250	49218	12305	3692	49380	55370	51226
4	7	49	20.4	11000	6350	15000	10900	1.2	.333	50957	16986	5095	47407	59450	49955
5	9	81	12.35	18250	10500	12900	—3300	1.0	.200	55438	11088	3326	48478	60982	50141
6	11	121	8.26	27,200	15700	7450	—34400	1.2	—.167	49245	—8220	—2466	7215	45135	5982
7	12.5	156	6.41	35,100	20200	0	—74500		.....	25000	.....	.....	—39176	25000	—39176

Metal Joining wit

By D. C. Dilley

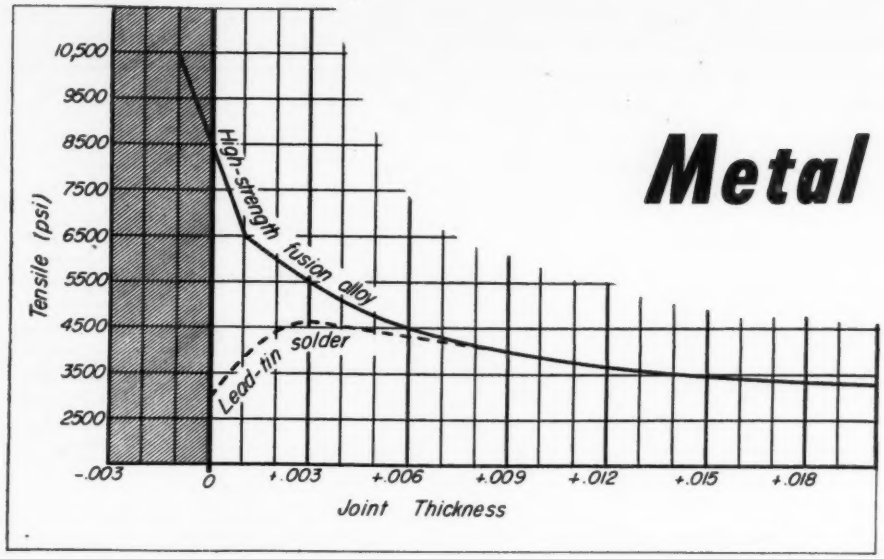
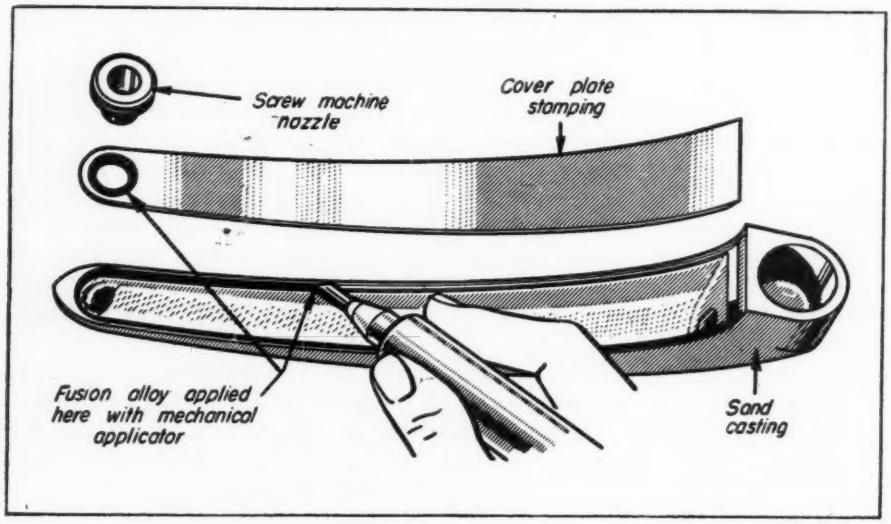
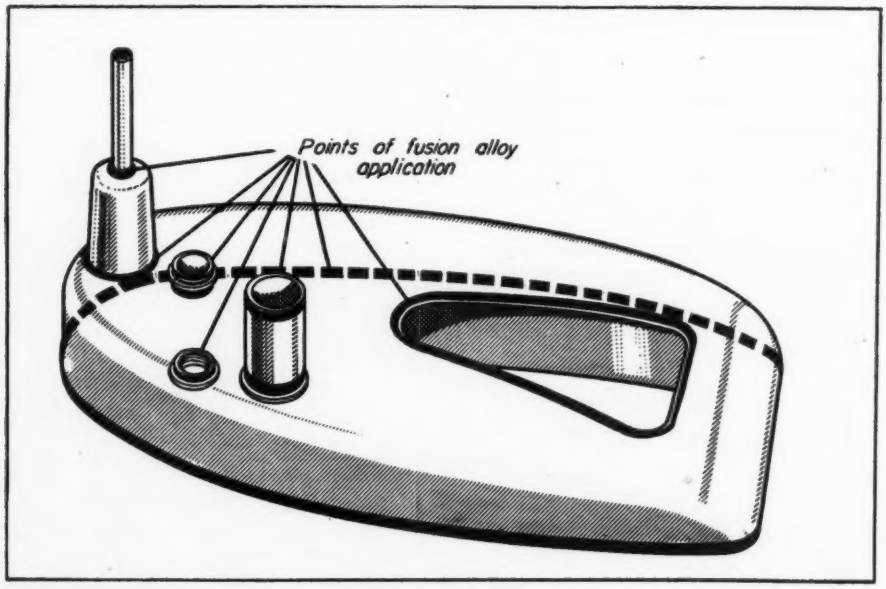


Fig. 1—Above—Curve illustrates how strength of joint made with paste type fusion alloy increases with decrease in joint thickness

Fig. 2—Below—Former design of this faucet spout utilized a one-piece yellow brass casting. Because coring difficulties gave rise to excessive rejects, built-up design was adopted





NEWCOMERS in the metal-joining field are a series of alloys in paste form which offer strong, clean joints which can be produced at temperatures generally lower than those required in other methods. These pastes can be applied by brush, by dipping, by roller in a manner similar to printing, or by pressure gun.

Fusion bonding with these alloys is performed without controlled atmospheres, and the process is free from toxic fumes and splatter. Except where additional fill is required, the amount of alloy often is one-third or less that required by other methods. This is due to the fact that no clearance is required between the pieces being joined. As the curves of *Fig.* 1 show, strength of paste type fusion-bonded joints increases with decreasing clearance between parts. In the resulting joint the paste alloy penetrates into the grain boundaries of the jointed metals.

These paste alloys, developed by The Fusion Engineering Co. of Cleveland and known as Micro-Film alloys, can be used for joining parts of aluminum, stainless steels, carbon and alloy steels, coppers, silver, brasses, tin, die-cast alloys and car-

Fig. 3—Left—Steam iron body is built up of thin copper stampings and screw machine parts. The eight steam-tight joints were produced simultaneously

...aste Type Fusion Alloys

bides. Stampings can be joined to sand castings, die castings to spinnings, screw-machine parts to stampings, etc. The following examples indicate how these alloys have been used to create better, low-cost assemblies than had been possible by previous methods.

FAUCET SPOUT: Approximate length of the modern faucet spout shown in *Fig.* 2 is nine inches. Former design utilized a sand casting of yellow brass and the coring presented difficult casting problems. Rejects were excessive, often being discovered after considerable work had been performed on the finished casting.

This spout was redesigned to comprise an assembly of three pieces: A casting, a stamping and a screw machine part. Main body of the spout was cast with the underside open. The underside cover plate was stamped and a hole punched in proper position to receive a screw machined nozzle. Fusion alloy was applied with a mechanical applicator to the groove where casting and stamping were to be joined, and at the same time at the point where the screw machine nozzle was to be joined to the stamping.

Heating was in a rotary core oven capable of maintaining a temperature of 550 F. The fusion alloys flowed into the joints and after burnishing and plating, the product was comparable in utility and appearance to the single casting employed in the original design.

STEAM IRON: The domestic steam iron body shown in *Fig.* 3 was built up of five stampings and drawings, plus three screw machine products. Joints had to be steam-tight and assembly had to be made quickly and easily. The stampings were of extremely thin copper alloy.

The dark bands in the illustration show points where fusion bonding alloys were applied in paste form prior to heating. With suitable fixtures the entire assembly was completed in one operation, eight steam or water-tight joints being formed simultaneously. It is interesting to note that in this application many of the joint areas were inaccessible during assembly prior to heating.

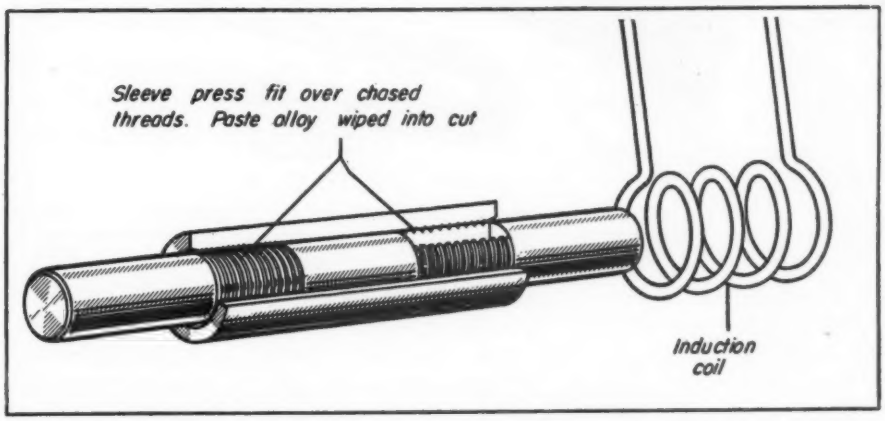
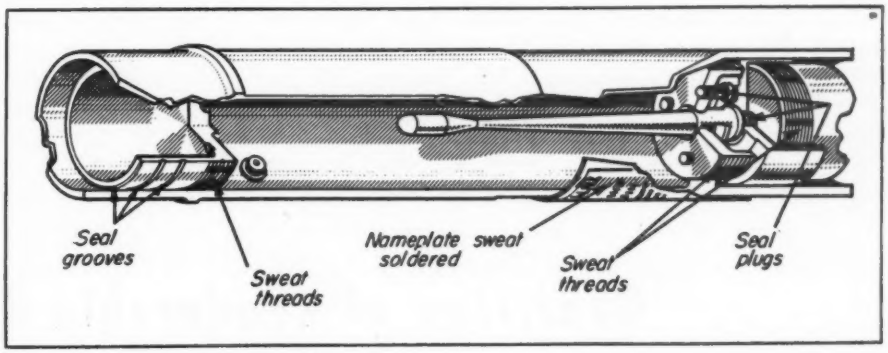


Fig. 4—Above—Use of paste type alloys permitted brazing sleeve to shaft and hardening surface of sleeve in single operation

Fig. 5—Below—Paste type alloys used as indicated in this aircraft shock-absorber strut produced hydrostatically tight assembly free of distortion



HARDENED SHAFT SLEEVE (*Fig.* 4): The problem in this case was to join a metal sleeve to a shaft of different composition and harden the outside surface of the sleeve at same time.

Light threads were chased on the shaft to a depth of about 0.005-inch. Alloy paste was applied to the shaft and the collar was pressed on. Next the entire assembly was heated in an induction coil to proper hardening temperature, and water quenched. Results were a hardened collar joined to the shaft—hardening and joining being performed simultaneously.

AIRCRAFT SHOCK ABSORBER STRUT: In producing the shock strut shown in *Fig.* 5 it was necessary to make it hydrostatically tight. Because of the thin wall sections extreme care had to be used in the final assembly since the slightest distortion from heating

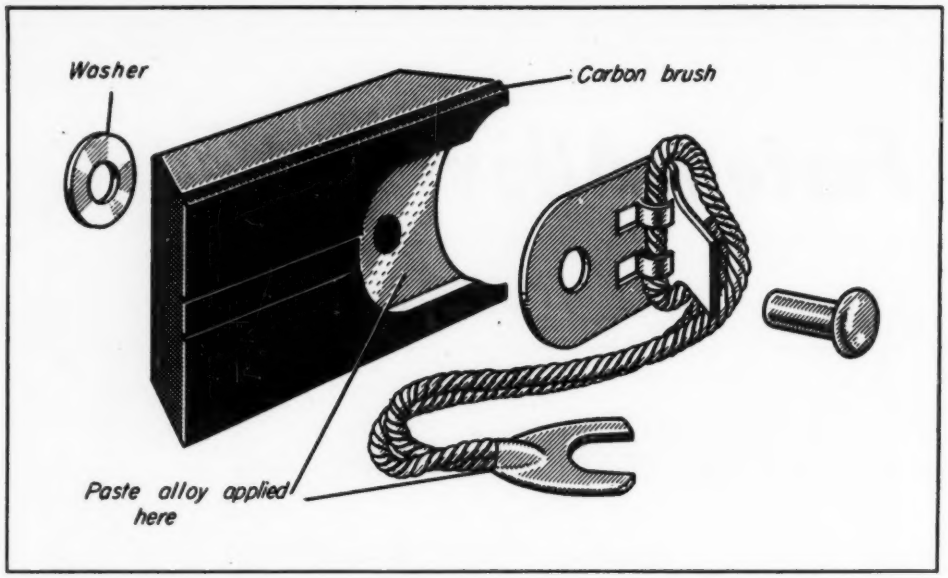


Fig. 6—Left—Strong joint having good electrical properties was required between lead clip and carbon brush. After copper plating brush pocket, clip was coated with paste alloy and fused to brush by heat applied with resistance spot brazer

would affect the fit between piston and cylinder.

Alloy paste was sweated into the threads during the heating operation. Assembly was made with parts preheated in order to minimize distortion and temperatures were low enough to prevent warpage. As an extra precaution, additional alloy was introduced through the seal grooves. It was also found that the nameplate could be easily sweated to the cylinder at the same time without the use of drilled holes, rivets, etc.

COMMUTATOR BRUSH: The carbon brush shown in *Fig.* 6 was copper plated in the region where the flexible lead is to be attached. The problem was to attach the lead clip to the brush with a strong joint having maximum electrical properties.

To insure the required strength, specifications called for a rivet, as shown. The lead clip was first coated with fusion alloy paste and pressed into the brush. The rivet and closing washer on the rear face of the brush were likewise coated with the alloy paste. After the rivet was made fast in the conventional manner, the brush assembly was quickly heated by a resistance spot brazer to fuse the paste alloy. The resulting joint was not only to specification electrically, but, since a high pH alloy was used, no subsequent cleaning was required to remove flux residue. When pulled under test, the carbon failed without rupturing the joint.

These examples illustrate the use of alloys melting from 300 to 1800 F. Electrical instruments, carbide-tipped tools, typewriters, and fishing lures, made from ferrous and nonferrous alloys, or from aluminum, are typical of the products now being successfully joined by this method.

Qualities of Leadership Defined

LEADERS are recognized by their success in being followed—and they are followed because they apply sound principles in their human relations. Sound human relations require intelligence, tolerance, understanding, and the sincere application of the Golden Rule. He who wants to lead must remember that most individuals want to help and be helped.

The real leader perseveres. He will admit error, but in admitting error he will continually push forward to attain a definite objective. He has a keen desire to work, finds pleasure in his work, and radiates this joy to those whom he contacts. The leader merits and enjoys the confidence of others. He carries with him a dignified simplicity in his aspirations to arrive at great heights. He will be kind in his thoughts and kind in his actions, and will retain this power of kindness against all odds.

One who would seek leadership or retain leadership must have a keen understanding of the obligation of his duty. He will be firm to keep faith with those who have placed upon his shoulders the responsibility for carrying out a given task. The leader must possess the rare virtue that goes hand in hand with kindness—patience. Patience must be a part of his life because impatience tears down and destroys much of the good work which has been accomplished. He will appreciate the joy of originating and will show appreciation for the efforts of others who originate, no matter how small their contribution may be toward bringing about better social and economic conditions. The leader will encourage constant thought for improved ways of doing things and for originating new things to be done. The leader will help others develop a strength of will and character that will enable them to withstand disillusionments and disappointments. Finally, he who would lead must be imbued with the idea of service based on the simple truth, said by the greatest of all salesmen, "But he that is greatest among you, shall be your servant." *From a paper presented at the SAE National Production Meeting in Cleveland, by J. E. Gulick, general manager, Tire Div., B. F. Goodrich Co.*

MACHINE Editorial DESIGN

Standardization Aids Design

Alert engineers responsible for the design of machines have long recognized that correct application of standards and standard practices is one of their most effective tools. They have taken full advantage of the opportunities offered them through the adoption of engineering standards to effect a high degree of simplification in design as well as untold savings in time and cost of manufacture. Had it not been for the utilization of the standards currently available, today's high peak of production could not possibly have been reached.

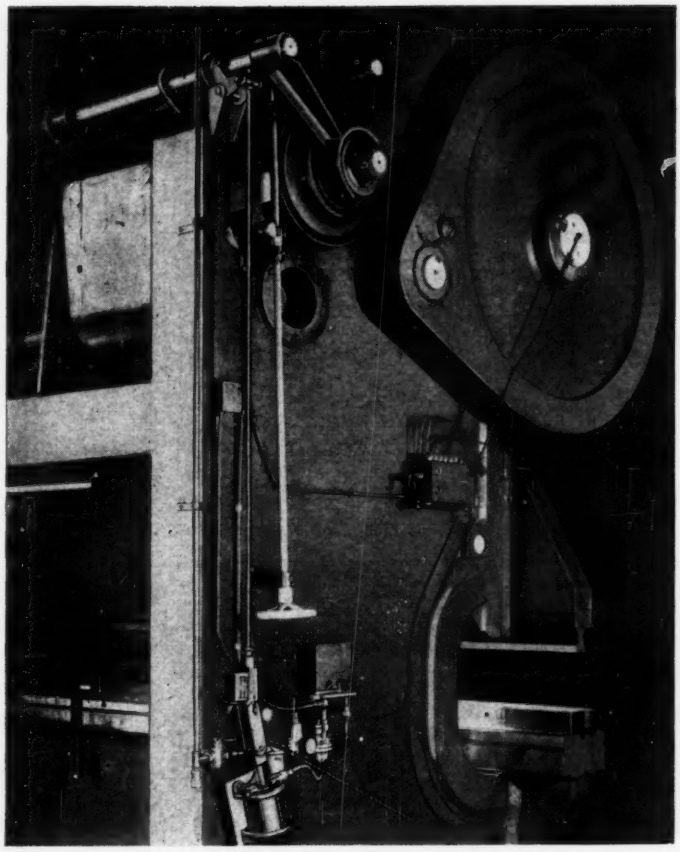
A feeling has existed in the minds of a relatively small number of engineers, however, that too strict an adherence to some engineering standards tends to retard rather than to advance development and design. Such a condition is conveivable in a few remote instances, but even in these cases the fault probably rests with the development of certain items in the particular standards being used rather than with the standards as a whole.

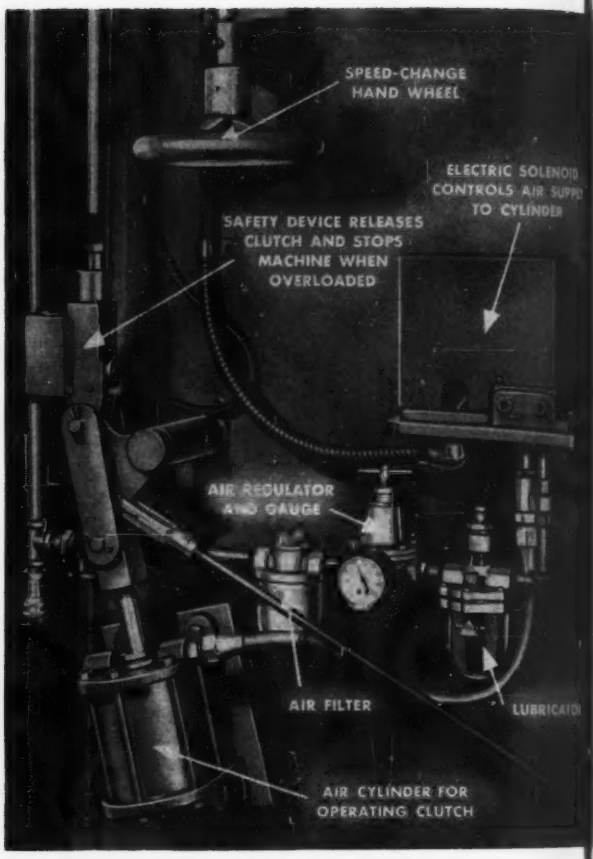
That the need for standards is constantly increasing is evidenced by the fact that at the last annual meeting of the American Standards Association it was pointed out that 169 new standards had been approved during the previous twelve months—a much larger total than in the preceding year. To these should be added the new internal standards developed within companies for use in those cases in which national standards have not yet been promulgated, or to supplement existing standards. Other standards or codes of standard practice also are being established by manufacturers' organizations or groups, a pertinent example of which are the hydraulic standards for industrial equipment presented in this issue.

Great credit is due the joint industries conference which compiled the new hydraulic standards. Tremendous increase in the application of machine hydraulics has made the standards virtually essential, and they unquestionably will be of far-reaching assistance to the designer in his work. It is to be hoped that inasmuch as they apply only to industrial machinery—aircraft, for instance, not being included—they will provide the basis for the formulation of a combined set of hydraulic standards acceptable not only to all types of manufacturers but also to all government agencies interested in the application of hydraulic equipment in machines. Thus would the standards, from their fine beginning, be of even greater ultimate value to the profession and to the nation.

L. E. Jermy

DESIGNS OF THE




SPEED-CHANGE HAND WHEEL
ELECTRIC SOLENOID CONTROLS AIR SUPPLY TO CYLINDER
SAFETY DEVICE RELEASES CLUTCH AND STOPS MACHINE WHEN OVERLOADED
AIR REGULATOR AND GAUGE
AIR FILTER
LUBRICATOR
AIR CYLINDER FOR OPERATING CLUTCH

Air-Electric Clutch Control

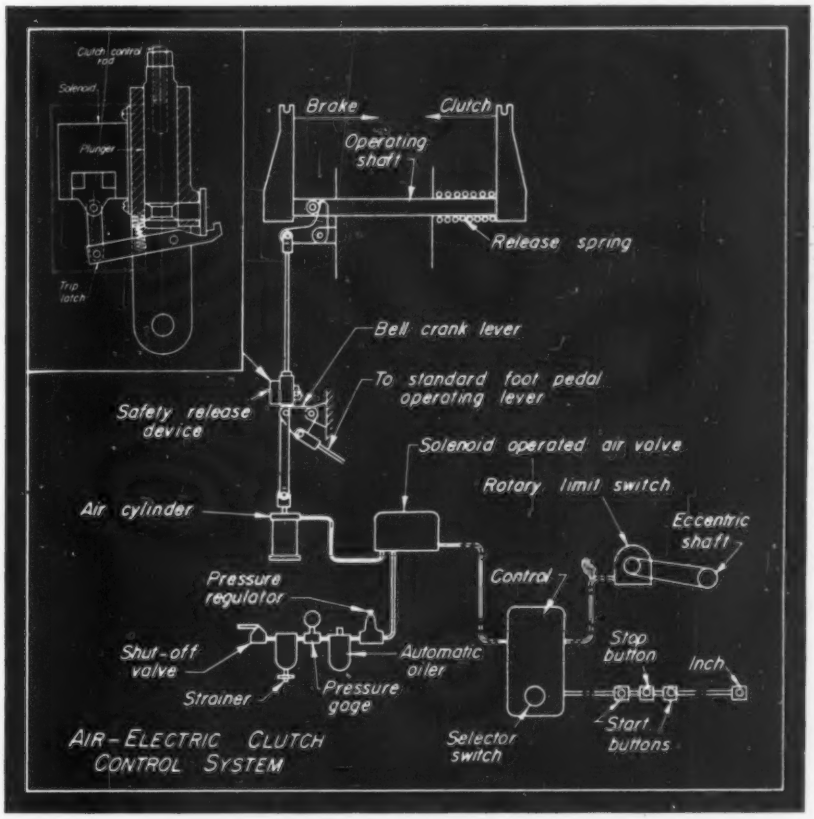
PROVIDING maximum operator protection and several operating cycles, a recently developed air-electric control system for press application is shown in the photograph above. As the schematic (top, next page) shows, the operator's start and stop signals energize the solenoid controlled air valve through a selector switch. An air cylinder actuates a toggle link, pulling down on the clutch control rod and engaging the clutch by means of a pivoted crank fastened to the upper end of the control rod. The brake and clutch are both operated from the same shaft. When the clutch is disengaged, a spring on the operating shaft automatically sets the brake. The photograph above, right, shows a close-up view of the air cylinder and component parts.

The safety release device (shown in inset, next page) disengages the clutch when the load on the press exceeds a predetermined amount. Limit switches are actuated by the load indicator, energizing the solenoid in the safety release device. The solenoid actuates the trip latch, allowing the upward pull on the clutch control rod plunger to force the retaining pin out of its slot in the plunger. This breaks the connection between the air cylinder and clutch shaft, releasing the clutch and immediately setting the brake.

Choice of any one of four cycles is available through positioning of the selector switch. In the "long" operating cycle, all start buttons (four if arranged for two man operation, and two if arranged for one man operation) must be depressed and held down until the dies close. The buttons may then be released, the slide stopping at the top of the stroke. Releasing one or more of the start buttons while the dies are closing stops the press immediately. The "short" cycle provides for release of all start buttons as soon as the press has been started, the slide automatically stopping at the start of the next stroke. The "continuous" is similar to "short" cycle operation with the exception that once the press has been

started it does not stop after each stroke. With the selector at the "inch" position all start buttons are automatically disconnected and the press will operate only while the "inch" button is held down.

The "safety nonrepeat" feature makes it impossible for the press to run through more than one cycle by holding down the start buttons in either "long" or "short" operation. Various arrangements of start buttons, either foot or hand operated, are available. Manufacturer: Cleveland Crane & Engineering Co., Wickliffe, Ohio.

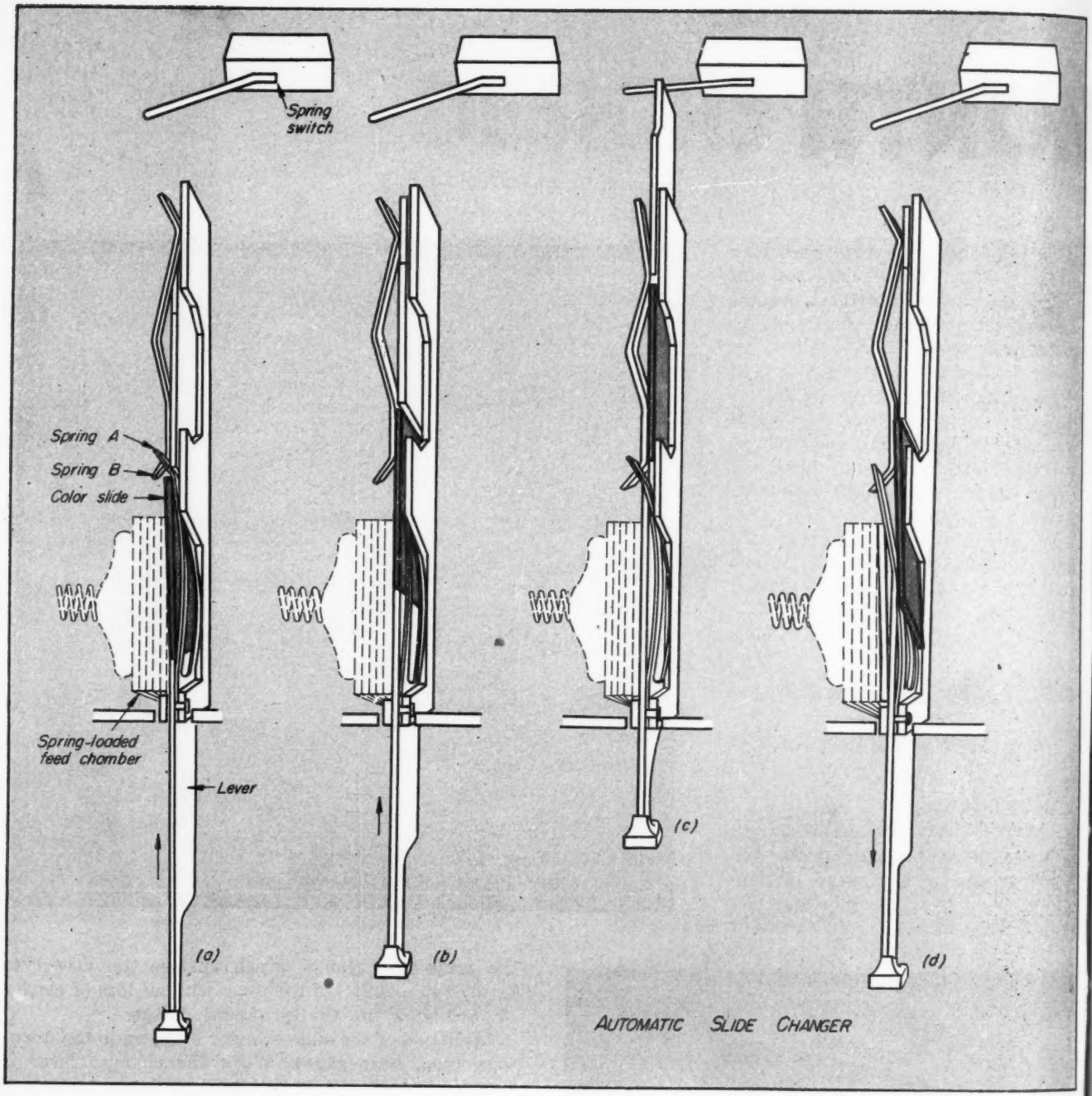


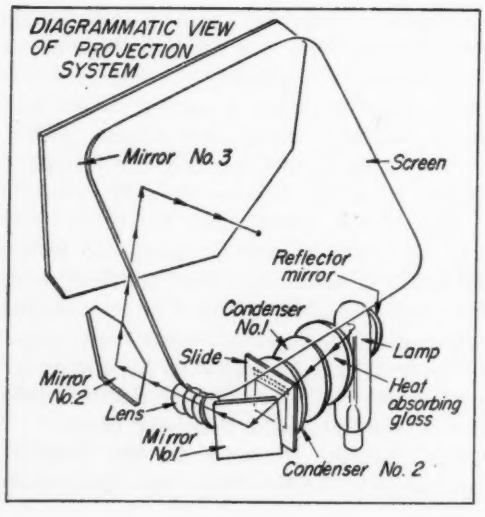
Portable Photographic Slide Viewer

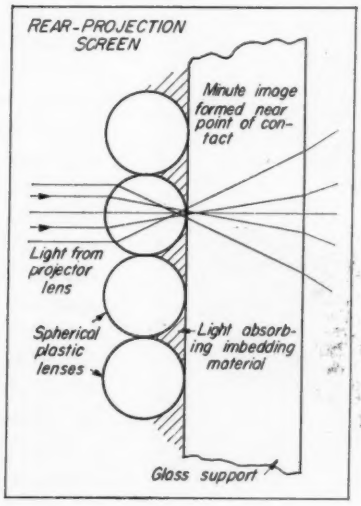
INCORPORATED in the table model slide viewer shown in the photograph are an automatic slide changer and a new type of rear-projection screen. Another feature is the projection system which enables the viewer to be used in a fully lighted room without loss of clarity or brilliance in the projected image.

Operation of the slide changer is shown in the drawings (top, next page). When the changer lever is pushed in, spring *A* is flattened, as shown in view *a*, and the slide in the feed trough is forced under spring *B*, as indicated in view *b*. Further movement of the changer lever permits spring *A* to snap back to its original position, spring *B* holding the slide in position for projection (see view *c*). When the changer lever is pushed all the way in it closes a spring switch and turns on the projection lamp. To change slides, the changer lever is withdrawn, pulling the last slide out under spring *B*, which forces the slide over on the ejection side of the mechanism (view *d*), and into the slide changer's receiving chamber where spring tension holds it in place. Simultaneously with ejection of the projected slide, the next slide is moved into the slide changer for projection when the changer lever is again pushed in. As many as 75 cardboard or 30 double-glass slides, intermixed if desired, may be placed in the spring-loaded feed chamber.

The projection system is shown in the diagram (bottom, next page). Light from the projection





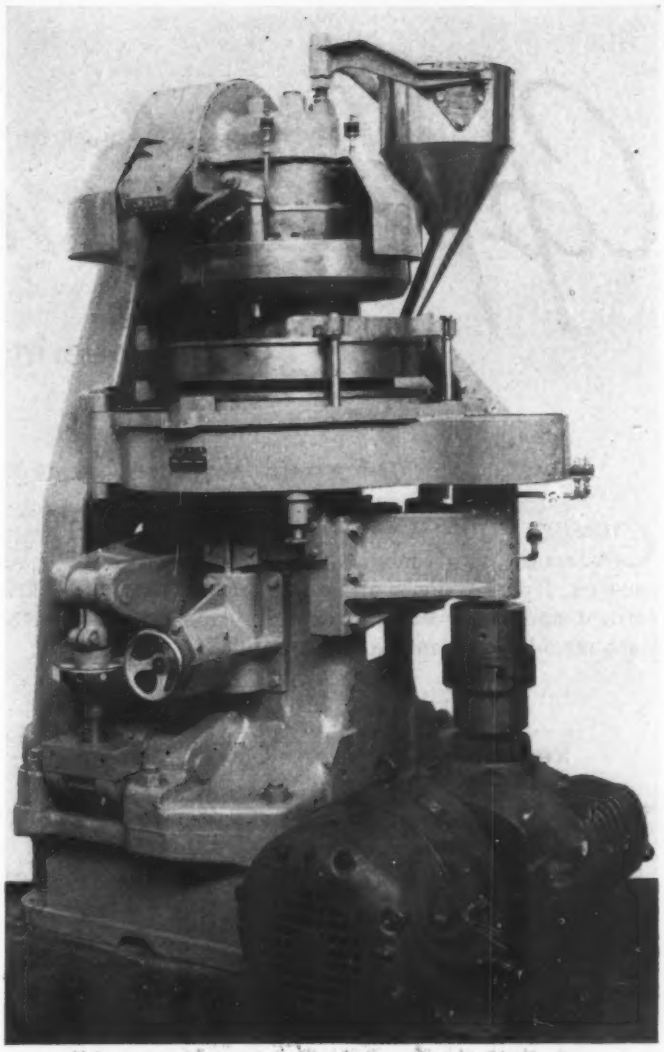


lamp is reflected by a spherical mirror located behind the lamp and passes through a heat absorbing glass and two condenser lenses before it passes through the slide. The image of the slide is projected onto a mirror, then through a focusing lens to a second mirror, from which it is reflected to a third mirror and onto the screen. This screen differs from the conventional projection system screen in that instead of being white it is black. The drawing, left, shows how thousands of small plastic

spherical lenses are imbedded in a light-absorbing material on the back side of the glass support of the screen. The point of contact between one of these beads and the glass support forms a small dot or pinhole. These beads not only act as small projection lenses by gathering the light striking them and projecting it through the screen, but they also increase the effective light-passing area of the screen by collecting the projected light striking the screen on all sides of the beads and directing that light through the holes. In addition to increasing the brightness of the image, beading the rear surface of the screen results in the light coming through each lens in a cone. This cone subtends several degrees from the normal to the screen, allowing the image to be seen over an appreciable viewing angle without distortion. The extremely close spacing of the plastic beads minimizes the grain of the screen. Manufacturer: Eastman Kodak Co., Rochester, N. Y.

High Speed Powder Metal Press

CAPABLE of producing 91 powder metal parts per minute, the 30-ton rotary press shown at right was recently introduced for high speed production work. Equipped with a 12-station head, the machine will handle parts up to 2½ inches in diameter with a die fill of 4¼ inches. Flanged parts can also be produced by the use of a mechanism that moves the upper pressure roll up or down in a vertical plane when used in conjunction with a special cam arrangement. A 20-hp constant speed motor powers the press, applying pressure from both above and below simultaneously. Other features include a pressure equalizer and excess pressure release, double lifting cams for the upper punches, core rod attachments, and lower punch bushings. Manufacturer: F. J. Stokes Machine Co., Philadelphia.



Checks Piston Pins Automatically

USING air jets for gaging diameters, the automatic piston pin checking and segregating machine, right, handles 2400 pins per hour. Parts are loaded into a chute which has a "nogo" gaging fixture to prevent oversize pins from being accepted. They are then gravity fed into a locating vee where a motor-driven arm pushes a pin into an air ring. There the pin is checked for average diameter by jets mounted in the air ring which operate "Airlectric" heads. These heads in turn actuate solenoids which open and close doors to the sorting chutes. The part is then moved on through the air ring by the next pin to be checked, is carried along by a traveling mechanism until it reaches the proper trap, and is then permitted to drop into the correct tray. Manufacturer: Sheffield Corp., Dayton, Ohio.

Applications

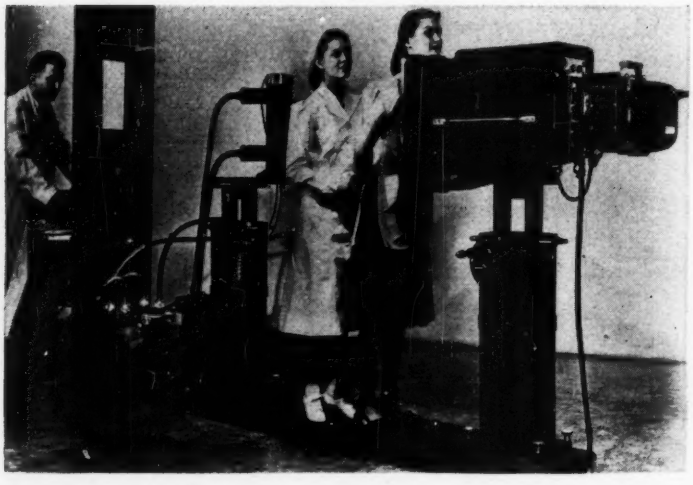
of engineering parts, materials and processes

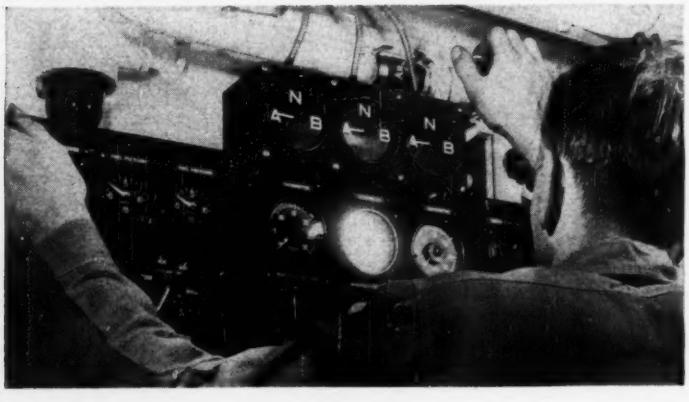
Contour-Formed Mounts

COMBINATION of processes has been used to make the aircraft engine mount ring shown at right. The blank, sheared from ¼-inch aluminum sheet, was press-brake formed and then contour formed to a circle in the Cyril Bath machine seen in the background.

Reduces Weight of X-Ray Machine

WEIGHT of this portable X-ray machine built by North American Philips Co. below, has been decreased by 300 pounds through the use of magnesium castings. Possessing good shock resistance as well as excellent specific elasticity, magnesium is well suited to use in this precision machine which must withstand rough handling.





Hydraulics Signaling System

HYDRAULIC telegraph system, left, for signaling between bridge and engine room in the Navy's PT boats requires no external power source. The unit, known as the Liquidometer transmitter, consists of a dual hydraulic system joining a bridge telegraph to a repeater dial in the engine room. Action of the helmsman in moving the telegraph lever is reflected in the repeater instrument.

Hydraulic Standards for Industrial Equipment

PRESENTED IN THIS DATA SHEET are the recently approved Hydraulic Standards for Industrial Equipment compiled by the Joint Industry Conference (J.I.C.) of six co-operating groups of manufacturers. The standards apply to all types of hydraulically actuated industrial equipment, including but not confined to machine tools. Aircraft practice is not included. Although these standards, like any others, are subject to future modification the editors of MACHINE DESIGN feel that their publication in full at this time will serve a useful purpose in stimulating thinking, promoting the adoption of more uniform practices, and leading to worthwhile improvements in the standards themselves

General

HO.1—*a.* The purpose of these Standards is to provide detailed specifications for the application of hydraulics to industrial equipment which will promote safety of personnel, uninterrupted production, long life of the machine, equipment or tool, and will not limit or inhibit advancement in the art of hydraulics.

b. Where new developments in hydraulics will perform in a manner equal to these specifications they will be considered as equivalent.

HO.2—These Standards are proposed to cover hydraulic equipment in general applications. Any exceptions to these Standards will be as agreed between supplier and purchaser.

HO.3—The purchaser shall specify such additional details as are needed to meet unusual operating conditions.

HO.4—*a.* The builder shall submit necessary performance specifications on hydraulic equipment, including symbolic or schematic diagram, for purchaser's approval. Final diagrams and text shall conform to the hydraulic equipment shipped.

b. Part numbers shall be provided for all types of hydraulic equipment. Seal manufacturers catalog number shall also be provided where sealing devices are commercially available.

HO.5—All hydraulic equipment shall be installed and used in accordance with the recommendations of the manufacturers thereof.

HO.6—Hydraulic and lubricating systems shall be separated or in accordance with the following provisions:

a. In no case shall hydraulic fluid be used to lubricate ways, slides, or any exposed surface or mechanism.

b. When separation is impracticable, combinations utilizing lubricating systems are permissible, only if unit pressure is low, operation is infrequent or of an auxiliary nature.

c. When systems are combined means shall be provided to remove contamination.

d. Whenever hydraulic and lubricating systems are combined complete information shall be given on the hydraulic system diagram.

HO.7—*a.* Under normal conditions of continuous operation the pump inlet temperature of the fluid shall not exceed 130 F.

b. In no case shall a sensitive temperature measuring device placed anywhere in the hydraulic circuit indicate a local temperature above 250 F. This paragraph applies to petroleum base fluids only. For fluids other than petroleum base fluids operating temperatures shall be as agreed upon between supplier and purchaser.

HO.8—All industrial equipment at the completion of assembly and before shipment shall be given complete performance tests to determine conformity with equipment specifications. There shall be no evidence of external leakage.

HO.9—A pressure time diagram taken under operating conditions including data concerning maximum rate of pressure rise, maximum surge pressure and surge frequency shall be furnished when specified on purchase order.

HO.10—Hydraulic equipment shall be of proved engineering design, fabricated of material of accepted quality and with work-man-like finish; and must be capable of performing efficient, reliable service in the installation for which it is provided.

Diagrams

Circuit Diagrams

H1.1.1—*a.* SYMBOLIC DIAGRAM: A drawing or drawings showing by means of approved standard symbols each and every piece of hydraulic apparatus, including all interconnecting lines.*

b. SCHEMATIC DIAGRAM: A drawing or draw-

The Joint Industry Conference consists of the following co-operating groups: Automotive Joint Industry Conference; Hydraulic Equipment Manufacturers; Hydraulic Press Manufacturers; National Machine Tool Builders' Association; Resistance Welder Manufacturers Association; and Tubing and Fitting Manufacturers.

* Size of interconnecting lines and characteristics of electrical equipment also shall be shown. Data, or text, or both shall describe the function for each phase of the complete cycle.

ings showing the functional construction of all valves, controls, and actuating mechanisms including all interconnecting lines.*

H1.1.2—SYMBOLIC DIAGRAMS:

a. Symbolic diagrams shall be furnished with industrial equipment (Schematic diagram acceptable).

b. A copy of the diagram shall be firmly attached to the inside cover of the hydraulic control enclosure or hydraulic diagram enclosure, either by adhesive material or by permanent data pocket or clip.

H1.1.3—SCHEMATIC DIAGRAMS:

a. When requested on purchase order, a schematic diagram shall be supplied.

b. When requested on purchase order, a schematic diagram shall show and describe the flow paths for each phase of each complete cycle.

c. When requested on purchase order, a schematic diagram shall show in color only those portions of the circuit which are functioning at each single phase of the cycle (see H1.4).

H1.1.4—ALL DIAGRAMS:

a. The descriptive text shall explain the function of all components of the circuit and the sequence of the operations. Either the diagram or descriptive text shall include, in addition to the sequence of operations, the characteristics of any electrical equipment use, and the following:

1. All hydraulic equipment shall be identified. When possible, catalog number and manufacturer's name shall be shown.
2. Recommended size of piping.
3. Diameters of pistons and rods, length of stroke and estimated required force of cylinders when other than maximum pressure is applied.
4. Time of cycle, when pertinent (for example, time range of cycle exclusive of loading).
5. Operating pressures.
6. Horsepower and revolutions per minute of each pump drive.
7. Pump speed and delivery in gallons per minute.
8. Reservoir capacity.
9. Recommended oil viscosity range.
10. Displacement, speed range and torque rating of each hydraulic motor.
11. Data, or text, or both shall show operations performed and characteristics of any actuating electrical equipment used.

b. The related text and the diagram shall be shown on the same or facing pages.

c. All diagrams shall be marked with serial number or with the purchaser's order number of that equipment, or, if none, with some other symbol which will identify the particular apparatus to which the diagram applies.

d. Requested number of copies of all diagrams required and texts shall be forwarded by mail or in person to an individual delegated by the purchaser not later than the date on which the shipment is made.

Standard Symbols

INTRODUCTION: Standard symbols indicate the nature and function of component parts in the circuit.
Symbolic symbols show the function. Supplementary symbols show the method of operation such as manual, electrical, hydraulic, etc. Function and method of operation are indicated by combining basic and supplementary symbols.
The symbols may be adapted to any type of hydraulic system. They may be used equally well for each piece of hydraulic apparatus in basis diagram as well as for showing flow paths in schematic diagrams.
The Standard Symbols are shown on pages 158 and 159.

Standard Fluid Legend Code

H1.4.1—STANDARD COLOR CODE:

Pressure	Red
Return	Blue
Trapped oil, or restricted return	Yellow
Intake and drain	Green

H1.4.2—SINGLE COLOR CODE: Symbolic cross hatching can be used to identify hydraulic function.† Where areas will permit ready identification substitution of cross hatching for color code will be in agreement with purchaser.

Diagram Size

H1.5.1—All diagrams shall be on 8½-in. x 11-in. file size sheets, or in multiples thereof, if folded. The diagram shall be of such size as to clearly disclose all components of the system.

Hydraulic Controls

Definition of Controls

a. Manual controls are those actuated by the operator, regardless of means.

b. Automatic controls are those actuated in response to the cycle of the equipment.

Protection

H2.3.1—Over-pressure protection shall be provided on the discharge side of all pumps. The adjustments on the pressure control units shall be accessible. Provision shall be made to permit locking enclosure or compartment in which pressure control unit is mounted, or protective locking type adjustments shall be provided to prevent tampering.

H2.3.2—All pressure controls shall be so constructed as not to be adjustable outside nominal working ranges, and shall be marked to indicate nominal minimum and maximum pressures.

H2.3.3—Where there is more than one pump on the equipment and possible damage may be caused by the stoppage of any one pump, interlocks shall be provided so that the stoppage of one pump in the circuit will not endanger or damage the equipment. Where the starting of the pump in an improper sequence can result in damage, interlocks shall be provided to insure the starting of all pumps in proper sequence.

H2.3.4—All necessary valves for removal of any pump while other pumps are still in operation shall be provided.

H2.3.5—Where pumps are operating in series, pressure protection shall be placed between pumps in series in order to protect any pump from carrying the entire load (This will not apply to centrifugal pumps in series). All conductors between pumps shall be of adequate strength for the intended pressure. (See also Recommended Practice for steel tubing. Page 163.—Ed.)

H2.3.6—Where loss of working pressure on discharge side of pump or pumps may result in damage to equipment, loss of accuracy, or danger of injury to personnel, means shall be provided to prevent operation under those conditions. When lubricating pumps are used as set forth in paragraph HO.6 *b* the same conditions shall prevail.

Controls, Circuit

H2.4.1—When shifting from rapid traverse to feed, the starting position of the feed rate shall be maintained consistent with the type of service intended.

H2.4.2—Hydraulic circuits shall be so designed as to minimize the generation of heat and to conserve power.

H2.4.4—In each electrically controlled circuit, in case of solenoid failure, means shall be provided to prevent damage to equipment, loss of accuracy, and danger of injury to personnel.

H2.4.5—Neutral valve positioning controls shall be independent of and unaffected by adjustment of other controls.

H2.4.6—Feed and cycle limiting devices on mass production equipment where feed is preset shall be adjustable and so designed that the feed distance can be set and sealed (Except where adjustment is necessary to compensate for tool wear).

H2.4.7—*a.* POSITIVE POSITION STOPS shall be so designed that the dwell time will not be affected when positive position stops are reset.

b. POSITIVE SAFETY STOPS for machine slides

(*Continued on Page* 160)

*Size of interconnecting lines and characteristics of electrical equipment also shall be shown. Data, or text, or both shall describe the function for each phase of the complete cycle.

† Standard symbols will be worked out and published at the earliest possible date.

Glossary of terms used in the hydraulic specifications

Automatic Controls: Those actuated in response to the cycle of the equipment.

Back Connected: Where piping connections are on normally unexposed surfaces of hydraulic components.

Channel: A fluid passage the length of which is large with respect to its cross-section dimension.

Clarifier: A device for the removal of deleterious solids and which assists in maintaining the chemical stability of hydraulic fluid.

Compartment: A space within the base, frame, or column of the equipment. (See also Enclosure).

Cylinder: A linear motion device in which the thrust or force is proportional to the effective cross sectional area and to the pressure drop.

Cylinder, Single Acting: A cylinder in which the fluid force can be applied in only one direction.

PLUNGER TYPE: A cylinder in which the internal element is of a single diameter, and upon which the seal applied is of the contracting type.

PISTON TYPE: A cylinder in which the internal element is of one or more diameters, and the seal is of the expanding type.

Cylinder, Double Acting: A cylinder in which fluid force can be applied in either direction.

SINGLE END ROD: A cylinder with a rod extending from one end.

DOUBLE END ROD: A cylinder with two rods, one extending from each end.

Enclosure: A housing for hydraulic apparatus. (See also Compartment).

Filter: A device for the removal of solids from a fluid, wherein the resistance to motion of such solids is in a tortuous path.

Front Connected: Where piping connections are normally on exposed surfaces of hydraulic components.

Hydraulic Panel:

1. HYDRAULIC MOUNTING PANEL: A plate on which may be mounted a number of hydraulic components.
2. HYDRAULIC CONTROL PANEL: A grouping of hydraulic control units mounted to form one assembly on a plate, or in a casting, and having a single mounting surface.

Line: A tube, pipe, or hose which acts as a conductor of hydraulic fluid.

Line, Drain: A line returning leakage oil independently to the reservoir or vented manifold. If returning to reservoir, it shall have reservoir symbol; and if to a vented manifold, it shall have a vented manifold symbol (a single line inverted T).

Line, Exhaust: A return line which carries power or control actuating fluid back to the reservoir.

Line, Joining: A line which crosses or connects with another line on a diagram and which also connects hydraulically in construction. The junction of the lines shall be indicated by a dot which must be at least three times the width of the line.

Line, Passing: A line which crosses another line on a diagram, but which does not connect hydraulically in construction. Line on a diagram to have no swell at crossing.

Line, Pilot: A line which acts as a conductor of control actuating fluid.

Line, Working: A line which acts as a conductor of power actuating fluid.

Manual Controls: Those actuated by the operator, regardless of means.

Mass Production: For a model run, or an indefinite period of time.

Motors and Cylinders: Devices for converting hydraulic energy into mechanical energy.

Motor, Oscillating: A motor producing a maximum angular rotative movement of less than 360 degrees in either direction.

Motor, Rotary: A motor producing rotary motion having torque output proportional to the displacement per revolution and to the pressure drop between intake and exhaust ports.

Motor, Rotary, Fixed Displacement: A rotary motor in which the displacement per revolution is not adjustable.

Motor, Rotary, Variable Displacement: A rotary motor in which the displacement per revolution is adjustable.

Passage: A machined or cored connection which lies within or passes through a hydraulic component, and which acts as a conductor of hydraulic fluid.

Phase of Cycle:

0. NEUTRAL.
1. RAPID ADVANCE: The approach of the tools or workpiece to the feed position.
2. FEED: The portion of the cycle where work is performed on the workpiece.
3. DWELL: The portion of the cycle where the feed rate or pressure stroke is stopped.
4. RAPID RETURN: The return of the tools or workpiece to the cycle starting position.

Port: An opening at a surface of a component; e. g., terminus of a passage. May be internal or external.

Port, Valve: A controllable opening between passages; i.e., one which can be closed, opened, or modulated.

Positive Position Stop: A structural member which definitely limits the working motion at a desired position.

Positive Safety Stop: A fixed structural member which confines maximum travel within the design limits of the machine or equipment.

Pressure-Time Diagram: A graphical presentation of pressure in pounds per square inch plotted against time for a complete cycle of the equipment.

Pump: Device for converting mechanical energy into hydraulic energy.

FIXED DISPLACEMENT: A pump which delivers a relatively constant volume of fluid per cycle.

VARIABLE DISPLACEMENT: A pump in which the volume per cycle can be varied.

Restriction: A device which produces a deliberate pressure drop or resistance in a line or passage by means of a reduced cross-sectional area.

CHOKE: A restriction, the length of which is relatively large with respect to its cross-section dimensions.

ORIFICE: A restriction, the length of which is relatively small with respect to its cross-section dimensions.

Restriction Choke (Fixed Viscous): A nonadjustable resistor in which the pressure drop or rate of flow is not entirely independent of the viscosity.

Restriction Orifice (Fixed Nonviscous): A nonadjustable resistor in which the pressure drop or rate of flow is relatively independent of viscosity.

Schematic Diagram: A drawing or drawings showing the functional construction of all valves, controls, and actuating mechanisms.

Sealing Device: A part or assembly of parts used to prevent leakage between two or more other parts.

Strainer: A device for the removal of solids from a fluid wherein the resistance to motion of such solids is in a straight line.

Subplate (Back Plate): An auxiliary mounting for a hydraulic component to which piping connections are mounted.

Surge: A transient rise of hydraulic pressure in the circuit.

Symbolic Diagram: A drawing or drawings showing by means of approved standard symbols each and every piece of hydraulic apparatus including all interconnecting lines.

Trip: A mechanical element for the actuation of a position control.

Unit Production: One piece per setup.

ENGINEERING DATA SHEET

STANDARD SYMBOLS

LINES

Line, working	Line, pilot (L > 20W)	Line, drain (L < 5W)	Line, flexible
Connector (dot to be 3x width of associated line)	Direction of flow	Line, passing	Line, joining (tee, cross, etc) (dot to be 3 x W)
Reservoir (fluid tank)	Line, to reservoir Above fluid level Below fluid level	Manifold, vented	Plug or plugged connection
Testing station (gage connection) T S	Power take-off P O T	Restriction, choke, fixed viscous	Restriction, orifice, fixed nonviscous

PUMPS

Pump, single, fixed displacement PF	Pump, single, variable displacement PV

MOTORS AND CYLINDERS

Motor, rotary, fixed displacement MF	Motor, rotary, variable displacement MV	Motor, oscillating MO
Cylinder single acting Plunger type Piston type	Cylinder double acting Single end rod Double end rod	

MISCELLANEOUS UNITS

Motor, drive, electric ELEC. MTR.	Heat exchanger HE	Intensifier INT.	Accumulator ACC
Filter FILT	Strainer STRN	Pressure switch PR SW	Pressure gage
Spring	Shaft, rotating	Component enclosure	

VALVES

Valve, check	Valve, restriction, choke Variable viscous
Valve, restriction, orifice Variable nonviscous V	Valve, basic symbol (insert model no. for special valves) Method of indicating internal flow

VALVE EXAMPLES

Valve, manual shut off	Valve, maximum pressure	Valve, relief remotely operated	Valve, sequence directly operated
Valve, pressure reducing	Valve, flow control pressure compensated (PC) Viscous Non-viscous (PC)	Valve, shut off 2 position 2 connection	Valve, directional 2 position 3 connection
Valve, directional 2 position – 3 connection	Valve, directional 3 position – 4 connection Open center	Valve, directional 3 position – 4 connection Closed center	

METHODS OF OPERATION

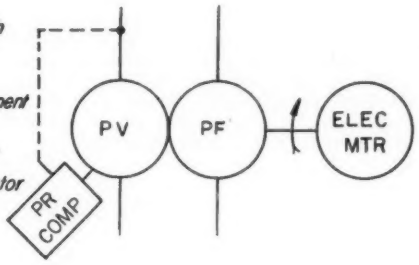
Control, basic symbol	Control, centrifugal — CENT	Control, compensator — COMP
Control, compensator pressure — PR COMP	Control, compensator temperature — TEMP COMP	Control, cylinder — CYL
Control, detent — DET	Control, manual — MAN	Control, mechanical — MECH
Control, motor, electric — ELEC MTR	Control, motor, hydraulic — HYD MTR	Control, pilot, hydraulic — HYD
Control, pilot, air — AIR	Control, servo — SERVO	Control, solenoid — SOL
Control, solenoid hyd. pilot operated — SOL HYD	Control, thermal — THERM	Control, pilot hyd. differential area — HYD DIFF (S, L)

EXAMPLES OF COMBINATIONS

Pump, double—with electric motor

one fixed displacement

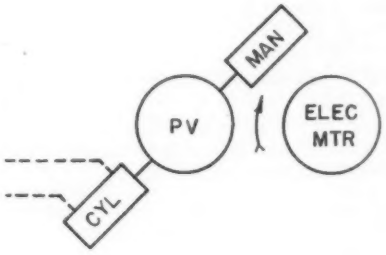
one variable displacement with pressure compensator



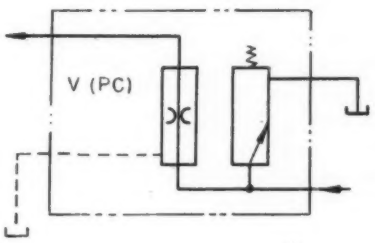
Pump, single- with electric motor

variable displacement

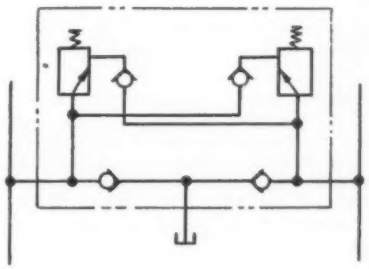
hand wheel and cylinder control



Valve, flow control and maximum pressure with compensator

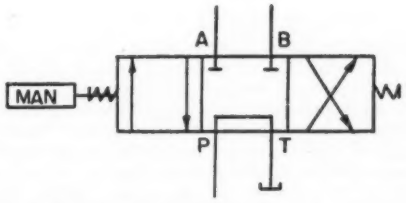


Valve, replenishing unit



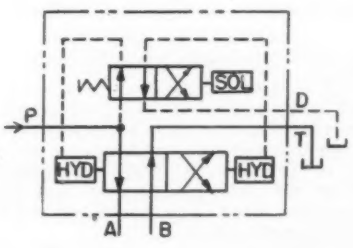
Valve, 4 way three position- spring centered manual control

P→T, cyl. port blocked in center position



(NOTE: symbol shown in center position)

Valve, 4 way two position spring offset solenoid control pilot operated



shall be adjustable, but shall have provision for replacement.

H2.4.8—On industrial equipment operated at elevated temperatures or with toxic materials, the circuits shall be so designed and located that they can be serviced while equipment is operating under these conditions.

H2.4.9—Hydraulic circuits shall be so designed, constructed and applied that unnecessary surge pressures are minimized at their source, and equipment be of adequate strength to withstand all operating pressures and surge pressures.

H2.4.10—Hydraulic circuits shall be so designed as to prevent uncontrolled movement of the equipment operating devices in all phases of the equipment cycle, including the idling phase.

H2.4.11—Hydraulic feed circuits shall be so designed and applied as to maintain consistent feed rates irrespective of reverse force or intermittent loads.

H2.4.12—Circuits having two or more heads or slides shall provide means to insure all units operating in proper relation regardless of load variation.

H2.4.13—Hydraulic circuits in mass production equipment shall be so designed and applied that load variations, and changes in fluid temperature ranges if automatically controlled, shall not cause variations in the cycle time of more than plus or minus 3, 5, 10, or 15 per cent, of the specified basic cycle provided voltage, frequency, and manual operations do not change. Purchaser shall advise the percentage of degree of accuracy required.

Enclosures and Compartments

H2.5.1—For the purpose of these standards, an *enclosure* means a housing for hydraulic apparatus; a *compartment* means a space within the base, frame, or column of the equipment.

H2.5.2—Automatic controls that might be damaged by exposure to dirt or dust or stock moving equipment shall be adequately protected and readily accessible.

H2.5.4—Doors and other parts of compartments for automatic controls shall be sheet steel not less than 14 USS gage, or cast metal not less than ¼-in. thick. Compartments for automatic controls shall be completely isolated from coolant, cutting or lubricating oil, chips, and dirt. All automatic control devices shall be mounted in one enclosure where possible and advisable.

H2.5.6—All enclosures for control devices, where practicable, shall have hinged covers which swing horizontally, or self-supporting sliding covers, and shall be held closed with screws which require the use of a tool (wrench or screw driver) to remove.

H2.5.7—The arrangement of enclosures for mounting controls shall permit ample room for proper maintenance.

Mounting

H2.6.1—*a.* ALL CONTROLS: All controls shall be mounted so as to minimize the possibility of damage from floor trucks or other stock moving equipment. They shall be protected against falling objects and operator's accidental movements. No controls shall be placed in high heat areas or in locations subject to any harmful conditions.

b. MANUAL CONTROLS: All manual controls shall be mounted within reach of the operator when in his normal operating position. Operator shall not be required to reach past revolving spindles or moving tools to reach manual controls (see Paragraph H7.1.2).

c. AUTOMATIC CONTROLS: Automatic controls shall be so located and protected as to prevent inadvertent operation.

H2.6.2—All hydraulic controls shall be mounted two feet or more above the bottom of the equipment providing such mounting will not impair the proper functioning of the circuit. This paragraph is written as a guide for all new machinery and equipment with the understanding that any deviation will be in accordance with Paragraph HO.2.

H2.6.3—Hydraulic control units shall be mounted in positions which provide complete accessibility to all units for maintenance without interfering with adjacent equipment.

H2.6.5—All industrial equipment shall incorporate an emergency stop control readily accessible from operator's normal position.

H2.6.6—Internal leakage must be drained to reservoir or vented manifold to prevent blocking of plunger type controls.

Pumps

Mounting of Pumps

H3.3.1—Pumps shall be mounted where they are readily accessible for maintenance and are adequately protected from damage.

Recommended Practice for New Design: Pumps shall not be mounted inside reservoir.

H3.3.3—When pumps and other apparatus are mounted inside the compartment or enclosure, sufficient space must be provided to assure complete accessibility to all units.

H3.3.4—Pump compartments must be clean and free of moisture and adequately vented to the exterior of the industrial equipment; all openings must be of sufficient height above the floor or be protected to such an extent that dirt, chips, coolant, etc., cannot enter, especially at such times as floor is swept or washed, or the equipment is cleaned. Unless other compartments meet the requirements of the pump compartment, there shall be no openings of any kind between the pump compartment and any other compartment.

If piping is run into the pump compartment from another compartment not meeting the pump compartment requirements, adequate seals shall be provided.

H3.3.5—Direct-coupled pumps must be properly aligned and securely mounted in such a manner as to assure alignment under normal operating conditions. When mounted inside column or base of equipment, foot mounted pumps and driving motors shall be mounted on a subbase of adequate strength which can be easily removed for servicing. The coupling must have adequate capacity to transmit the power required.

Pump Nameplates

H3.5.1—The following information shall be permanently indicated on each hydraulic pump:

a. Manufacturer's name and address
b. Manufacturer's identifying part number
c. Manufacturer's serial number if any.

A data sheet or catalog providing complete information on speed limits, pressure range, delivery, horsepower required, and viscosity range, shall be provided with the specifications.

H3.5.2—Whenever a pump is mounted so that its nameplate is not readily visible, a plate with duplicate information shall be provided where it can be readily seen. Such duplicate plate shall be marked "duplicate". The nameplate shall not be removed from the pump.

H3.5.3—The direction of rotation of each pump shall be clearly indicated on the pump where it can be readily seen.

Piping, Fittings, and Fluid Passages

(Piping includes all pipe, tubing, hose and fittings. Passages include all fluid conductors other than piping.)

H4.1.1—When requested by purchaser, a piping layout or photograph shall be furnished.

H4.1.3—Cross-sectional area of the piping shall be such that the flow capacity shall be sufficient on the suction side to prevent cavitation or starvation of the pump; and on the discharge side to prevent undue temperature rise or turbulence. Piping shall have adequate strength to withstand additional pressure, including surge pressure, within the safety limits of the entire circuit. A factor of safety of at least eight (8) over normal working pressure is suggested.

H4.1.4—*a.* Unnecessary surges shall be minimized at their sources, and adequate protective devices shall be

provided in piping circuits to protect piping and fittings against hydraulic impact.

b. The strength of the piping shall be adequate to withstand the maximum rate of surge pressure rise and the maximum surge peak pressure at the frequency developed by the cycling of the equipment operation.

H4.1.5—Piping leads in working circuits between actuating devices and feed control devices shall be so constructed as to rigidly confine a minimum volume of fluid, the purpose being to maintain constant controlled motion and restrict the effect of varying forces.

H4.1.6—*a.* Solderless piping connections shall be used.

b. The performance of any type of fitting used shall equal or exceed the requirements of AN-F 47.

c. When flared type fittings are used (which require a flaring tool), the angle of flare of tubing shall be 37 degrees from the centerline (74 degrees included angle) and flare dimensions shall be in accord with Army-Navy Design 10061.

d. When flared type fittings are used, the construction of the tube end (i.e., tubing end of connector body, nut, and sleeve when used) shall conform to joint Army-Navy Specification AN-F 366 for minimum performance requirements. Modifications may be used if performance conforms with the requirements of sentence *b*.

e. All pipe threads shall be Dryseal American (National) Standard Taper Pipe Threads (N.P.T.F.). (This is to become mandatory within six (6) months after the Hydraulic Standards for Industrial Equipment have been authorized for publication as approved standards.)

f. On installations in which fitting alignment is important, fittings with straight threads and which incorporate seals that seal with pressure may be used in place of pipe thread fittings.

Definition of Pressures

Low	0 to and including	200 psi
Medium	Above 200 to and including	500 psi
Medium High	Above 500 to and including	1200 psi
High	Above 1200 to and including	3000 psi
Extra High	Above 3000 psi	

Tubing and Fittings (See Recommended Practices on Page 163)

1. Tubing outside diameter sizes shall increase in 1/16-in. increments from ⅛ to and including ⅜-in., and in increments of ⅛-in. above ⅜-in. diameter up to and including 1 in. diameter and in ¼-in. increments above 1 in. diameter.

2. SAE 1010 dead soft, cold-drawn seamless steel tubing or equivalent and steel fittings shall be used.

3. Copper tubing or copper alloy fittings are not permissible with petroleum base fluids.

4. Fittings with restricted or stepped-up passages are not recommended.

H4.1.7—All piping, piping fittings, oil passages, reservoirs, cored holes or drilled holes must be free of burrs, or foreign matter which might cause damage to any hydraulic unit or contamination of the hydraulic fluid. Sharp edges shall be removed whenever they adversely affect the flow of hydraulic fluid.

H4.1.8—Whenever practicable, each piping run must be integral and continuous from one piece of apparatus to another. Piping runs must be removable without dismantling equipment components and without bending tubing or springing it in a manner to damage it. All rigid piping must be securely supported at frequent intervals to avoid vibration or movement. Points of contact or support on the outside of the piping shall be so designed that they will not result in damage to the piping. Piping shall not be welded to supports.

H4.1.9—If high or extra-high pressure piping above 1 in. is used, all connections shall be welded to steel flanges (all welded joints shall be stress relieved); each run shall be separate so that it can be removed by flanged connection using sealing devices that seal with pressure.

H4.2.4—In all portions of piping circuits above 150 psi operating pressure subject to movements in which flexible hose is required, the flexible rubber hose shall be fabricated to a pressure safety factor of at least five (5) with respect to the circuit in which it is used. Below 150 psi operating pressure, the use of plastic tubing shall be permissible. Flexible hose and plastic flexible tubing shall have vertical connections and shall have sufficient slack to avoid sharp flexing and straining. Horizontal connections are acceptable only if tubing is adequately supported.

H4.2.5—Multiple controls shall be mounted on a single plate within one enclosure, whenever possible, to eliminate excessive piping.

H4.2.6—Piping shall not be placed where it will interfere with the adjustment, repair or replacement of control or units, and piping connections shall be readily accessible for maintenane.

H4.2.7—Hydraulic units provided with external drain connections shall be drained independently to reservoir or to a vented manifold.

Identification of Piping

H4.4.1—Piping runs shall be identified to correspond with diagram.

H4.4.2—When construction of equipment necessitates shipment or moving in sections, thorough precautions shall be taken to adequately protect piping runs, whether they be left in place on one of the equipment sections or whether they are removed and separately boxed for shipment. All male threads shall be sleeved, and all piping openings or female threads equipped with suitable closures.

H4.4.3—Piping shall be so arranged that pressures may be tested at accessib'e locations, preferably at built-in checking stations which shall be indicated on diagrams.

Oil Reservoirs

(Separate or Integral with Machine)

H5.1—A hydraulic fluid reservoir shall be designed and constructed in such a manner as to prevent the entrance of any foreign matter, chips, coolant, lubricating or cutting oil, dirt, etc.

Fluid reservoirs should preferably be made separate and removable from equipment bases.

If cast, fluid reservoirs must be impregnated and sealed before assembling into equipment, with a sealer which is inert to hydraulic fluid. This must be tested by means of light petroleum solvent. If welded, welds must be tested with light petroleum solvent.

H5.1.1—The filler hole shall have a fine mesh screen, removable for cleaning and replacement, for straining the oil when filling, and shall be provided with a well fitted cap or cover permanently attached to the reservoir by some suitable means.

H5.1.2—A breather hole properly protected by an air cleaner shall be provided. The breather and filler must be separate and have sufficient capacity to permit rapid filling. The air cleaner must also be of sufficient capacity to maintain approximate atmospheric pressure at maximum circuit demands.

H5.2—The capacity of the fluid reservoir shall be sufficient to contain all of the fluid that will drain from the system back into the tank by gravity flow; maintain the fluid level within a reasonable working height during operating cycle; dissipate heat generated by normal operation of the equipment; allow air and foreign matter to separate out of the fluid.

H5.2.1—The reservoir shall be equipped with a flush mounted or protected fluid level indicator, preferably located near the filler cover. It shall be provided with markings to show high and low level with pump running and, where necessary, high level with pump stopped.

H5.3—Both the intake and return pipe shall be brought down below the minimum working fluid level so as not to cause cavitation or aeration.

H5.3.1—Suitable means shall be provided between the intake and return pipes to facilitate settling out of foreign matter in the fluid. The intake and return pipes

shall not be located in a manner which will interfere with the efficient operation of the hydraulic system.

H5.4—Ample and accessible provisions shall be made for complete cleaning of the reservoir.

H5.4.1—A well protected, accessible means shall be provided to empty the reservoir without spillage.

Heat Exchangers

H5.6—The heat exchanger provided, if required, shall be of proper design, and its application shall be subject to approval by purchaser.

H5.6.1—It shall be the responsibility of the purchaser to advise supplier incorporating the heat exchanger in the equipment of any unusual temperature that would affect the equipment operation.

H5.6.2—The heat exchanger must be readily accessible for removal, in the event servicing or repair is necessary.

H5.7—The heat exchanger shall be so arranged that it is not possible to develop excessive pressures in the event the heat exchanger should become clogged.

H5.7.1—Heat exchangers using media which are deleterious to the system's hydraulic fluid shall not be installed in fluid reservoirs.

H5.8—The make, size, capacity, and requirements of heat exchangers (including supply pipe sizes) shall be agreed to by the purchaser, and shall be clearly indicated on hydraulic diagrams.

H5.9—Accessible arrangements shall be provided for checking inlet and outlet temperatures of the hydraulic fluid if circulating type heat exchangers are used.

H5.10—When heat exchangers are used, they shall be equipped with suitable automatic thermal controls.

H5.11—Heat exchangers shall not use copper or copper alloys in contact with uninhibited petroleum base fluids.

Valves, Accessories, and Devices

Filters

H6.1.1—Means shall be provided for the continuous removal from the hydraulic fluid of materials which would be detrimental to the operation of the equipment.

H6.1.2—If the filter is of such design that it removes antifoam agents or other additives, the type and design of the filter shall be agreed upon between the supplier and purchaser.

H6.1.3—Self-cleaning filters shall be automatically actuated.

H6.1.4—Filters shall be of such construction that the filter medium can be removed for cleaning without disturbing the piping. When available, suitable means shall be provided to indicate when filter should be cleaned.

H6.1.5—When suction strainers are provided they shall be of ample size to strain more than double the intake capacity. Suitable means shall be provided to remove strainers without draining the reservoir.

H6.1.6—By-pass filter must be of sufficient capacity to filter the equivalent of all hydraulic fluid in the system within a period of eight (8) hours.

Sealing Devices:

H6.2.1—All sealing devices must be of such materials that they will not be adversely affected by hydraulic fluid. Natural rubber shall not be used with petroleum products.

H6.2.3—Sealing devices on reciprocating or rotating shafts shall prevent all leakage, except that required for lubrication of such devices under all working conditions without damaging shafts.

H6.2.4—Holes in sealing glands shall be of such size as to prevent undue extrusion of the sealing material.

Cylinders and Pistons

H6.3.1—Cylinder bores having fitted pistons shall be finished in a manner consistent with the type of service intended and shall be free of porosity or other defects.

H6.3.2—Cylinders shall be located in alignment with work slides, and shall be such that no side or radial load shall occur on piston rod or ram, unless other suitable provisions are made to take such loads.

H6.3.3—Ends of cylinders shall utilize sealing devices that do not leak under any working conditions. Sealing devices that seal with pressure are recommended. All cylinder external sealing devices shall be readily acessible for servicing.

H6.3.4—RECOMMENDED PRACTICE: Cylinders shall be separate, not cast integral with equipment, and in any event shall be accessible for servicing.

H6.3.5—Pistons utilizing sealing devices which are not integral with the piston shall have automotive step-cut rings or equivalent in performance. Exceptions are permissible only for applications where no leakage (other than required for lubrication) can be tolerated.

H6.3.6—Piston rods shall be adequately protected by proper location, or by installation of cleaning and sealing devices. Where conditions require, material and hardness of piston rods shall be such as to prevent scoring.

H6.3.7—*a.* When pistons are assembled to rod they shall be positively locked.

b. Flow of fluid into nonrotating cylinders shall be at extreme tops of bores or other means provided to prevent entrapment of air.

H6.3.8—When necessary, cushions shall be provided on cylinder ends.

H6.3.9—Wherever possible, cylinders shall be so used in the circuit or circuits as to subject the solid head ends of cylinders to high pressure, and not the gland or rod ends.

Valves

H6.4.1—Adjustable valves shall be arranged so that wires and seals can be used after final adjustments and locking.

H6.4.2—Valves shall not be mounted in reservoirs except in self-contained units. Where valves, including solenoid operated valves, are mounted in reservoirs, they shall be kept above high fluid level, so maintenance and adjustments can be made without draining fluid, except where the function of the valve requires same to be below the oil level.

H6.4.3—Gasket mounted valves shall be used whenever possible so their removal and replacement can be made without breaking individual fluid lines. Gaskets of the pressure sealing type are recommended. Gasket mounted valves shall have locating pins.

H6.4.5—The operation of a valve shall not produce detrimental surges in the hydraulic system.

H6.4.6—On vertical or inclined slides or rams means shall be provided to prevent rapid drop when hydraulic power is shut off.

H6.4.7—Solenoid operated valves shall be so designed, constructed, and installed as to eliminate destructive hammering.

Accumulators—(H6.5)

Safety

H7.1.1—Flexible lines shall be restrained or confined if their failure might constitute a hazard to personnel.

H7.1.2—Operator shall not be required to reach past revolving spindles, moving tools, or moving machine or equipment elements to reach manual controls.

H7.1.3—Hydraulic circuits incorporating accumulators shall be so interlocked as to vent or isolate accumulator fluid pressure when power is shut off. On circuit applications where accumulator pressure is isolated, full information shall be given on or near accumulator for proper servicing without injury to personnel.

Recommended Practices

Fittings

See H4.1.6 and H4.1.9

Steel Tubing

LOW PRESSURE. Steel tubing is recommended; plastics and aluminum tubings are permissible. Recommended minimum wall thickness for steel tubing is given in the accompanying table.

Recommended Minimum Wall Thickness for Steel Tubing

Tubing Diameter (in.)	Low and Medium * Pressure Wall Thickness (in.)	Medium High Pressure Wall Thickness (in.)	High Pressure Wall Thickness (in.)
⅜	0.035	0.035	0.060
½	0.035	0.035	0.075
⅝	0.035	0.049	0.095
¾	0.035	0.049	0.109
⅞	0.049	0.065	0.109
1	0.049	0.065	0.120
1¼-1½ incl.	0.065	0.095	*

* Above 1 inch, welded flange fittings or approved equivalent.

MEDIUM, MEDIUM-HIGH AND HIGH PRESSURES. Recommended minimum wall thicknesses for steel tubing are given in the accompanying table.

EXTRA HIGH PRESSURE. Specifications to be developed later.

Seamless Low Carbon Steel Tubing Annealed for Bending—Pressure Type

The following requirements are recommended practice but not mandatory under these standards.

1. SCOPE: This specification covers cold drawn and annealed seamless low carbon steel pressure tubing, for use as hydraulic lines.

2. HYDROSTATIC TEST: Tubing supplied under this specification shall have been tested at not less than 1,000 psi hydrostatic pressure, or higher in case working conditions require.

3. CLEANLINESS: It is desired that the inside of tubing shall be bright, clean, and free from grease, drawing compounds, oxide, scale, carbon deposits, and any contamination that cannot be readily removed by cleaning with alkaline cleaners or benzine.

4. CORROSION PROTECTION OF INTERIOR OF TUBES: The inside of the tubing having open ends shall be protected by a coating of clean oil to protect the interior of the tubing against corrosion during shipment and normal storage periods.

The corrosion preventive oil used shall be such that after extended storage periods, it can readily be removed with an alkaline cleaning solution or with benzine (Stoddard solvent).

‡ For tubes with an outside diameter ⅝-in. and less, and wall thickness 0.035-in. and less, a minimum elongation of 30 per cent is permitted.

5. QUALITY: Tubing shall be free from defects not in accordance with the best commercial practice.

6. CHEMICAL COMPOSITION: The tubing shall conform to the following chemical analysis:

Carbon, per cent	0.08—0.18
Manganese, per cent	0.30—0.60
Phosphorus, per cent	0.050 Max
Sulphur, per cent	0.055 Max

7. PHYSICAL PROPERTIES: The tubing shall conform to the following physical requirements:

Tensile Strength, max.	55,000 psi
Elongation in 2 inches, min.	40%‡

A three-inch section cut from the tubing shall not crack or show any flaws when flattened between parallel plates to three times the wall thickness of the tube.

A section of tube approximately 4 inches in length with the ends burred, shall withstand being expanded at one end over a polished tapered mandrel having an included angle of 60 degrees until the actual average inside diameter is increased 30 per cent.

The expansion test shall be made in a die in order to restrict the expansion of 30 per cent. The axis of the mandrel should be parallel to the axis of the tube during the expansion test.

8. PERMISSIBLE VARIATIONS: The diameter of the tubing shall not vary from that specified more than the following amounts:

Outside Diameter	Outside Diameter Plus (in.)	Outside Diameter Minus (in.)	Inside Diameter Plus (in.)	Inside Diameter Minus (in.)
Below 3½ to 1½ incl.	0.010	0.010	0.010	0.010
Below 1½ to 1, I. D., incl.	0.005	0.005	0.005	0.005
Below ½ I.D.	0.003	0.003		

When the outside diameter and wall thickness of ½ to 3½ in. OD tubes are specified, the outside diameter shall conform to the tolerances given in the above table, the wall thicknesses shall not vary from that specified more than plus or minus 10 per cent. When less than ½ in. I D, the outside diameter shall conform to the tolerances given in the above table. The wall thickness shall not vary from that specified more than plus or minus 15 per cent.

9. PICKLE BRITTLENESS: Tubing which is pickled to remove scale shall be treated to eliminate pickle brittleness.

10. LENGTH: The tubing shall be furnished in random 4 to 8-ft lengths unless otherwise specified on the purchase order. When the length is specified, the tolerance shall be minus 0, plus ⅛-in. unless otherwise specified. When cut to length, the ends shall be free from burrs.

11. PACKING: Material shall be packed according to the requirements of the transportation companies and of the purchaser.

12. MARKING: Each length of tubing 1½ in. OD and over shall be steel stamped with the manufacturer's identification mark at a distance of approximately 1½ in. from the end. This mark shall be permanent but not deep enough to injure the tube.

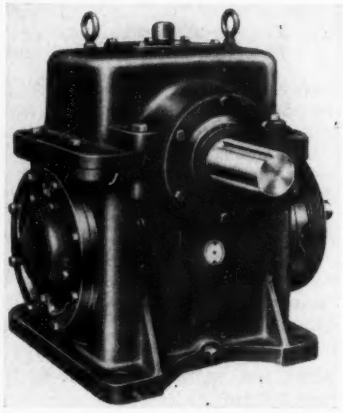
13. ALL BOXES AND BUNDLES SHALL BEAR THE FOLLOWING INFORMATION: Purchase order number, purchaser's specification number, manufacturer's name, size and gauge of material. In carloads, when shipped loose, shipping notices and talley sheets will show contents of car.

14. TERMS OF REJECTION: Material failing to comply with this specification may be rejected and returned for replacement or full credit, at option of purchaser.

new parts and materials

For additional information on these new developments see Page 271

Worm-Gear Speed Reducer



Line of enclosed worm drives is available in both horizontal and vertical types. Units have increased load-carrying capacity owing to a new gear-generating technique. They also have increased thermal capacity because of air-cooling of the oil reservoir, a high-velocity stream of cool air being passed through a duct immersed in the reservoir. The reducers are made in a wide range of standard ratios complying with the speed-ratio standards of the AGMA and NEMA. Manufacturer: Foote Bros. Gear and Machine Corp., 4545 Western Blvd., Chicago 9.

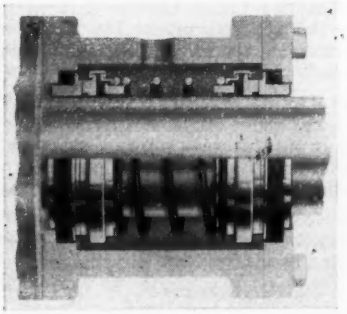
For additional information circle MD 1 on Page 271

Metallic High-Gloss Enamel

Baking enamel known as Chrome-Aluminum enamel has the appearance of polished chromium. The finish, which may be applied by spray, dip, brush or tumbling, has excellent flowing characteristics making it suitable for one-coat coverage of most surfaces. Manufacturer: Norfolk Paint & Varnish Co., Quincy 71, Mass.

For additional information circle MD 2 on Page 271

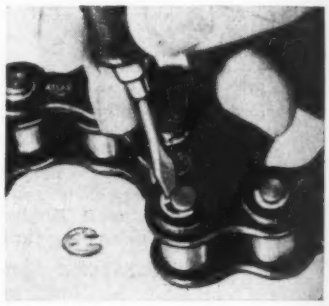
Rotating Shaft Vacuum-Pressure Seals



Designed for use on rotating shafts, Roto-Flex seals are suitable for use with water, oil or gasoline pumps, refrigeration compressors and speed reducers, etc. They may be used to seal against either vacuum or pressure. Units are of simple construction for ease of installation and incorporation into design. They are available in two styles. First is a single unit with one sealing element. Second is a double unit consisting of two rotating seals mounted on opposite ends of a compression spring. Both are designed for installations where the liquid being sealed cannot act as a lubricant or coolant and a separate liquid supply must be admitted to the seal chamber. Manufacturer: Gits Bros. Mfg. Co., 1868 S. Kilbourne Ave., Chicago 23.

For additional information circle MD 3 on Page 271

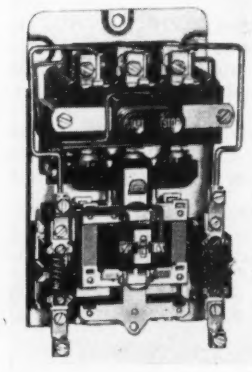
Roller Chain



New type roller chain, available in a range of sizes, uses snap retaining rings to secure roller pins. The rings are provided with three internal equally spaced teeth which maintain a high tension when the ring is sprung into a groove in the pin. Because of the construction, the ring turns in groove when struck rather than springing free; because of large cross-sectional area, it resists direct blows well. In the event of disassembly, the construction of the ring allows necessary spread for removal without taking a permanent set. Disassembly is possible with ordinary screwdriver. Manufacturer: Atlas Chain and Mfg. Co., Castor and Kensington Avenues, Philadelphia 24.

For additional information circle MD 4 on Page 271

Across-the-Line Starters

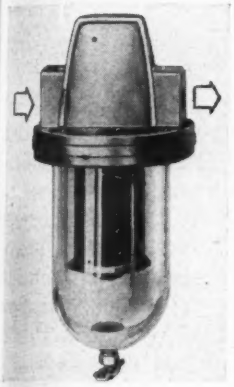


Bulletin 4112 Size 2, alternating-current magnetic starters are units of the across-the-line nonreversing type. They are solenoid-operated units intended for use with polyphase squirrel-cage induction motors as well as single-phase motors. These controls are available with open type construction for specialized application or with NEMA type I, general-purpose enclosure. They can be controlled by separate pilot devices or can be supplied with local control pushbuttons or selector switches. Some of

the features are high arc-interruption capacity, good thermal overload protection and built-in solderless connectors. The starters have a maximum enclosed rating of 25 hp, 440-550 volts, 3-phase, 60 cycles. Standard operating coils can be supplied for 110, 220, 440 and 550-volt, 25 or 60-cycle service. Dual-voltage coils can be furnished for 110/220 or 220/440-volt, 60-cycle service. Manufacturer: Ward Leonard Electric Co., Mount Vernon, N. Y.

For additional information circle MD 5 on Page 271

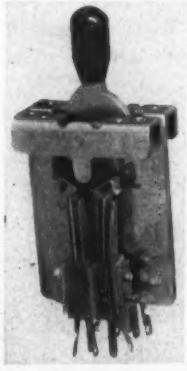
Compressed-Air Line Filters



Combination air filters are suitable for use in removing moisture, emulsified oil, grit, and undesirable impurities from compressed air lines. Air enters the bowl of the filter through spiral baffles which cause it to swirl in a circular path, throwing off impurities toward the wall of the bowl. Impurities that are not removed in this manner are extracted by a filter cartridge using a method known as edge filtration, the air passing between layers of the cartridge. Action of the filter is visible through large-capacity, transparent plastic bowl. Accumulation of liquids and sediment can be discharged by opening a petcock at the bottom of the unit. The cartridge is cleaned by closing the air supply line and permitting entrained compressed air to blow back. The units, designated as series RF air-line filters, are made in ⅜ and ½-inch sizes for straight-through connections and for working pressures up to 150 psi. Manufacturer: Hannifin Corp., 1101 S. Kilbourne Ave., Chicago 24.

For additional information circle MD 6 on Page 271

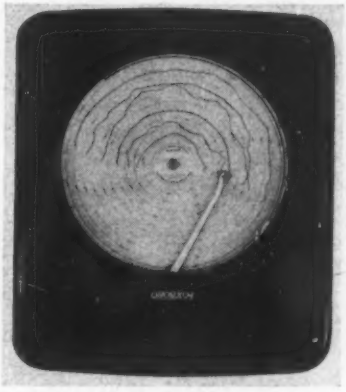
Subminiature Switches



Midget lever type switch, model MCT, has integral shielding and single-hole mounting. The shielding is said to be particularly valuable when intercircuit coupling is to be avoided; single-hole mounting facilitates installation. The switch is available in two types: The MCT-4 has behind-panel depth of 2 21/32 inches, including terminals; the MCT-1 has a behind-panel depth of 2½ inches. By the incorporation of stops, the switches can be provided with locking or nonlocking feature on one or both sides. Manufacturer: General Control Co., 1200 Soldiers Field Rd., Boston 34.

For additional information circle MD 7 on Page 271

Electronic Recorder



High speed electronic recorder makes from one to six different graphical records on a single circular chart. Instrument has a single measuring system, the six records being made by a positive-acting switching unit which automatically brings colored pens into recording position at six-second intervals. Innovation in the unit is the pen wheel which holds the six recording pens. These are selected in turn and held in recording position by a single pen arm. Record appears as dots so closely spaced as to appear like a continuous line. Manufacturer: Foxboro Co., Foxboro, Mass.

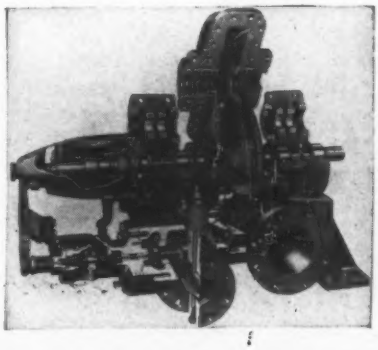
For additional information circle MD 8 on Page 271

All-Metal Flexible Joints

All metal Swivlex swivel joints are suitable for use with fluids at 1000 psi and 600 F. The units use stainless-steel bellows which serve as seep-proof self-adjusting springs to keep the metal-to-metal seat under preadjusted pressure. An internal equalizing chamber balances the pressures on each side of the seat, and ball bearings reduce friction to a minimum. The joint is therefore free turning and highly resistant to leaks. Standard sizes are made up to ½-inch ID. Applications include presses, vulcanizers and other hydraulic, pneumatic and steam equipment. Manufacturer: Seamlex Co. Inc., 41-23 24th St., Long Island City 1, N. Y.

For additional information circle MD 9 on Page 271

General-Purpose Turbines



General-purpose turbine line is made in types for operation with steam up to 1500 psi and 950 F. Ratings available are in the range 5 to 1500 horsepower. The units, trade named Typee, are recommended for use in driving pumps, fans, blowers, compressors, paper machinery, and small generators. They feature the use of combination felt and labyrinth bearing seals

and an overspeed trip which simultaneously closes the governor valve and a heavy butterfly valve. Basic turbine is equipped with a shaft type governor mounted in a sturdy casing that completely protects the governor assembly and shaft end. Units can be equipped with special wide-range governors where required. Three wheel sizes available include 16, 20 and 25-inch. These may be used with steam conditions up to 600 psi and 750 F. Equipped with heavy duty parts, the units are operable at 1500 psi and 950 F. Manufacturer: Westinghouse Electric Corp., P. O. Box 868, Pittsburgh 30.

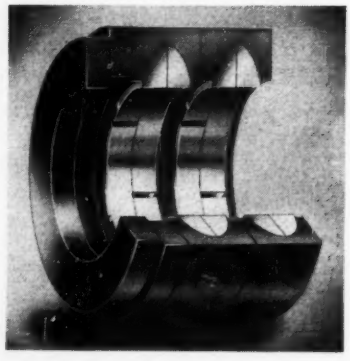
For additional information circle MD 10 on Page 271

Rubber-Plastic Materials

Blends of high-styrene copolymer and rubber are tough and light and have high impact resistance. The materials are made of various types of rubber and synthetics, the rubbers being used to plasticize the copolymers. Outstanding properties of the new family of materials include: Low water absorption, ease of post forming, good working characteristics, wide range of available physical properties and wide range of possible colors. The materials are designated Tuf-Lite. Manufacturer: Goodyear Tire & Rubber Co., Chemical Div., Akron 16.

For additional information circle MD 11 on Page 271

Hydraulic Packing

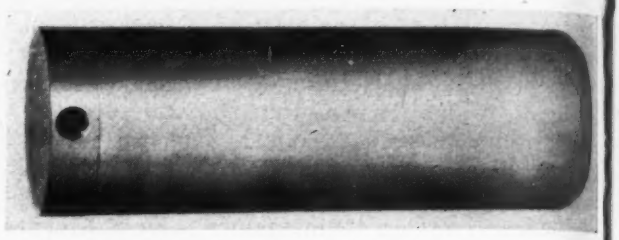


Hydraulic packing known as Monopak is suitable for use with many liquids and gases over a wide range of temperatures and pressures. It will seal oil, water, brine, weak acids and caustics, and refrigerants, and withstands temperatures from minus 50 to plus 300 F. It will resist pressures up to 3000 psi. The material is suitable for use with rods, packed pistons or grooved pistons and is effective on either reciprocating or rotating surfaces. Typical applications include pressures, struts, boosters, accumulators, pumps, valves, and compressors. Manufacturer: Hydraulic Accessories Co., 439 E. Fort St., Detroit 26.

For additional information circle MD 12 on Page 271

High-Pressure Filter Units

Filter assemblies using porous stainless-steel elements are designed for service at differential pressures up to 10,000 psi. Owing to the porosity of the filter element, high flow capacities are possible. Bodies of the units are either carbon or stainless



steel, the latter being made in the quick-opening type. Applications include hydraulic systems, compressor lines, etc. Manufacturer: Micro Metallic Corp., 193 Bradford St., Brooklyn 7, N. Y.

For additional information circle MD 13 on Page 271

Vibratory Bowl-Type Feeder

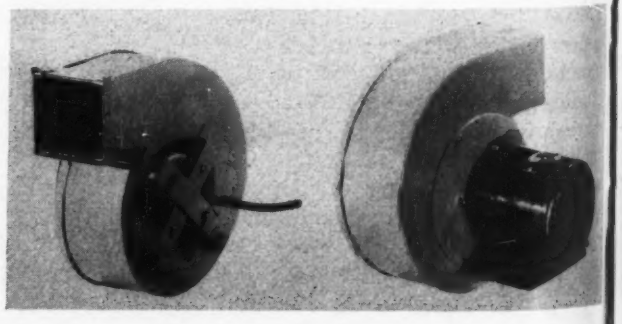
Electromagnetic vibration is used in the Syntron vibratory, vertical-bowl feeder to feed small parts or bulk powders. The feeder has a grooved track up the inside of the bowl wall. Vibration causes the parts or material to climb this rack at a uniform pace. In the case of parts, the progression, or feeding, can be made single file. The unit will feed at low velocities with speeds readily adjusted. Operation is from 110, 220 or 440-volt, ac power. Manufacturer: Syntron Co., 260 Lexington Ave., Homer City, Pa.

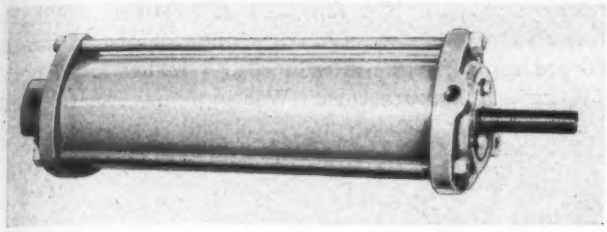
For additional information circle MD 14 on Page 271

Centrifugal Cooling Fans

Line of small centrifugal fans with wheel diameters of 5 inches provides a wide selection of capacities for many cooling and ventilating needs. These type 5CB fans can be supplied with a range of motor sizes, the motor size and speed being determined by the air flow required. Alternative concentric motor mounting results in compact unit of high efficiency. Brackets and mountings are available to meet a range of needs. Manufacturer: Johnson Fan & Blower Corp., 1319 W. Lake St., Chicago 7.

For additional information circle MD 15 on Page 271





Pneumatic Cylinder

Made in standard bore sizes from 1½ to 10 inches in diameter, the JLE pneumatic cylinder is available in any stroke specified. Top operating pressure is 110 psi. A low-cost type, the cylinder is made in five standard mountings. Manufacturer: James L. Entwistle Co., 43 Church St., Pawtucket, R. I.

For additional information circle MD 16 on Page 271

Wear-Resistant Metal Bushings

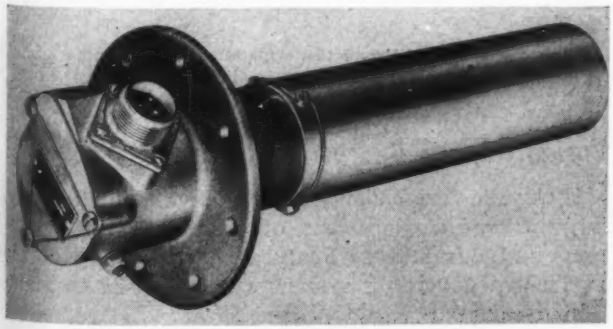
Line of bushings is made in Kennametal Grade K6, in 12 OD sizes ranging from ⅜ to 1 inch. The bushings are supplied rough molded with only sufficient stock provided for finish grinding. Length and inside diameter of the bushings are made to suit requirements of the user. Manufacturer: Kennametal Inc., Latrobe, Pa.

For additional information circle MD 17 on Page 271

Float Switch

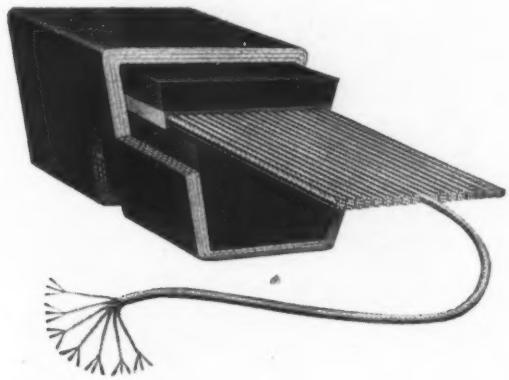
Float switch suitable for measuring all types of liquids features the use of a tube-enclosed float. The unit will not jam when subjected to shaking or vibration. Switch has a fully enclosed electrical circuit and is designed for minimum space requirements. It has a wide adjustment range and may be easily adjusted after installation. Rating is 10 amperes, mounting is from top, bottom or side. Manufacturer: Hydro-Aire Inc., 3000 Winona Ave., Burbank, Calif.

For additional information circle MD 18 on Page 271



Flexible Transmission Belts

Transmission belts of the V-type contain multiple strands of steel cable to provide high strength and long life. Type A belts are made in pitch lengths from 27 to 113 inches; type B in pitch lengths from 36.4 to 240 inches. Type C belts are made in pitch lengths from 53 to 240 inches. These transmission



elements, tradenamed Gripbelts, are said to have little stretch and give long life without loss of power and efficiency. Manufacturer: Browning Mfg. Co. Inc., Maysville, Ky.

For additional information circle MD 19 on Page 271

Small-Size Wheel Brake

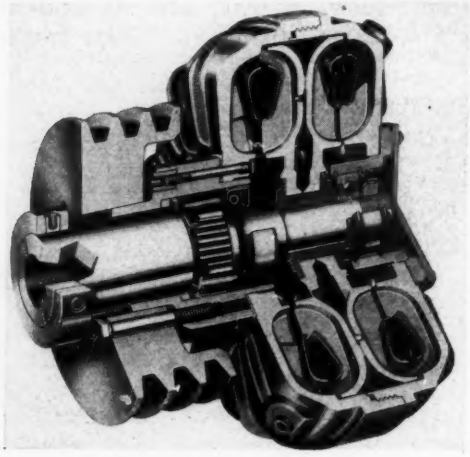
Adjustable brake for small wheeled units provides control for wheel capacities of 12, 14 or 16 inches OD. A single cap screw and locknut controls the internal expanding floating wedge, providing a graduated adjustment of brake shoe to drum surface. Split-type design allows easy demounting without disturbing bearing adjustment. Applications include power-driven industrial trucks, motorized scooters, etc. Manufacturer: Saginaw Products Corp., 68 Williamson Bldg., Saginaw, Mich.

For additional information circle MD 20 on Page 271

Hydraulic Power-Transmission Unit

A combined hydraulic coupling and V-belt sheave, the Hydro-Sheave Drive is provided with a sleeve which slides over the output shaft of any small electric motor or engine. Models are available in the horsepower range ¾ to 25. This drive unit reduces

installation length by obviating outboard bearings and flexible couplings and may be installed either on the motor shaft or on the primary shaft of the driven equipment. Sheaves are available to accommodate A, B and C-section belts in 1, 2, 3, 4 and 5 grooves, with



pitch diameters ranging from 4.2 to 12.4 inches. Typical applications include use on hoists, fans, blowers, machine tools, and equipment in the rubber, textile and food industries. Manufacturer: Twin Disk Clutch Co., Rockford, Ill.

For additional information circle MD 21 on Page 271

Automatic Voltage Stabilizers

Three new voltage stabilizers for use with 115-volt power are rated 15, 25 and 50 va. They provide a steady output of 115 volts (plus or minus one per cent for fixed unity power-factor loads) with input voltages ranging from 95 to 130 volts. Units have low case height and small size and are totally insulated. Manufacturer: General Electric Co., Schenectady 5, N. Y.

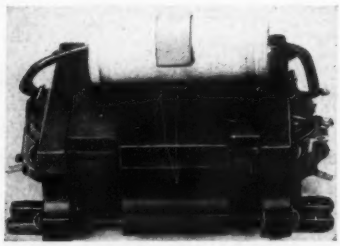
For additional information circle MD 22 on Page 271

Totally-Enclosed Nonventilated Motors

Silicone-insulated induction motors are totally enclosed and nonventilated, yet are of the same frame sizes, for equal ratings, as the open motors. Efficiency, power factor and torques are said to be also comparable to corresponding open motors. Size reduction by the use of silicone is indicated by the accompanying illustration in which the left-hand motor is insulated by the use of this material, while the right-hand motor of equal rating is class A insulated; frame sizes are 254 and 326 respectively. Weight reduction in the case of the 5-hp motors shown is 60 per cent. Height is reduced 3¼ inches and length 7¼ inches. Manufacturer: Westinghouse Electric Co., P. O. Box 868, Pittsburgh.

For additional information circle MD 23 on Page 271

Voltage Stabilizer

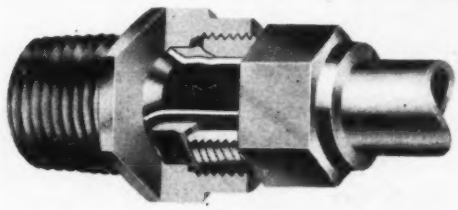


Voltage stabilizer is specifically designed for building into equipment where space and weight limitations are critical. This new 5-watt stabilizer is designed for operation at an input voltage of 95 to 130 volts, 60 cycles, single phase, the output being 120 volts stabilized to plus or minus ½-per cent. Manufacturer: Raytheon Mfg. Co., Waltham, Mass.

For additional information circle MD 24 on Page 271

Self-Flaring Tube Fitting

Leakproof, self-flaring tube fitting can be taken apart and reconnected many times without weakening the seal or damaging the flare. Unit, known as Uniflare, has two parts, the body and the nut. A thrust collar in the nut shears off and clamps onto the tube to form a perfectly sealed joint. Features of the fit-



tings are a 37-degree flare for maximum strength, inverted type joint to provide multiple sealing surfaces and close support to minimize vibration fatigue. Units are made in a complete range of sizes and types for use with copper, aluminum, Monel and annealed-steel tubing. Manufacturer: Scovill Mfg. Co., Waterbury 91, Conn.

For additional information circle MD 25 on Page 271

Variable-Speed Hydraulic Drives

Hydraulic variable-speed drives have high efficiency, large volume displacement, quiet nonpulsating operation, and durability. Feature of the drives is the Hydro-syn governing system which is used to maintain a constant predetermined speed. Desired speed

POWDER METALLURGY

Ledaloyl SELF LUBRICATING BRONZE

Low Cost, Self-Oiling, Long Life

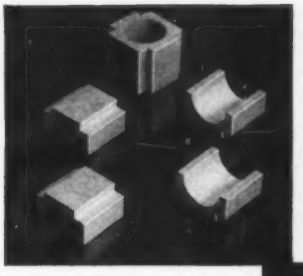
SLEEVE BEARINGS

MANUFACTURERS of many types of equipment can gain these advantages by using Johnson LEDALOYL Bronze for bearings and small parts. LEDALOYL is a product of powder metallurgy, manufactured by an exclusive process using pre-cast, pre-alloyed bearing bronze.

LEDALOYL bearings and parts are molded, under pressure, to required shape and size, thus eliminating expensive machining operations. When quantities of any size are used, the unit cost is surprisingly low.

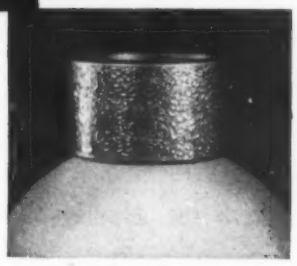
Another valuable feature is the self-lubricating action of LEDALOYL. Myriads of tiny, evenly spaced pores serve as miniature oil wells. When the bearing is in use the oil is metered to the shaft . . . when at rest the oil is absorbed by these pores. This action provides adequate lubrication at all times . . . preventing wear and in most cases eliminating the expense and bulk of other lubricating aids. Service records show long, trouble-free operation on many types of installations.

When you are looking for ways to reduce cost and to increase efficiency in your bearing applications . . . call on Johnson Bronze. We manufacture all types of Sleeve Bearings . . . we are ready to serve you now.



Low Cost

Odd shaped parts and self-aligning bearings when produced by conventional methods require considerable machine work, as compared with one forming operation in LEDALOYL.



Self-Oiling

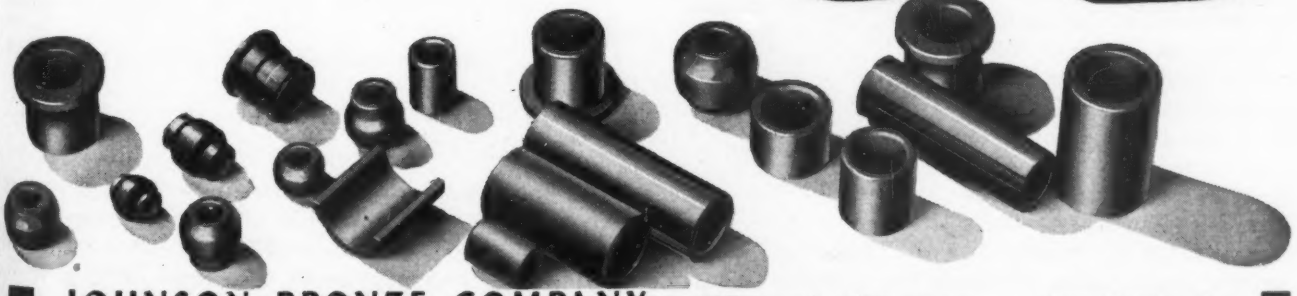
Place a LEDALOYL bearing on a lighted lamp bulb. Watch the oil sweat from the pores. Remove the bearing and the oil is reabsorbed. This illustrates the self-lubricating and oil retaining action.

Long Life

On a recent "break-down" test, LEDALOYL Bearings operated week after week on a schedule of nine hours a day for five days a week. The accumulated hours of test run represented a normal use of about five years, yet there was no sign of wear in the bearings or on the shaft.

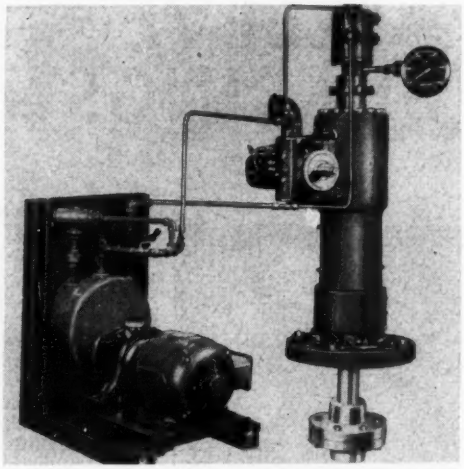
JOHNSON BRONZE
SLEEVE BEARING HEADQUARTERS

BRANCHES IN 20 INDUSTRIAL CENTERS



■ JOHNSON BRONZE COMPANY • 525 South Mill Street . . . New Castle, Pa. ■

is set by adjusting the position of a main circuit-control valve, using a graduated scale. The speed is then maintained by a separate hydraulic control circuit which operates on the constant pressure-differential principle, through a pilot valve. The system can be used to maintain any predetermined condition of speed or torque. Drives with speed ranges up to



5000 rpm, in ratings from fractional to over 125 hp are available. Manufacturer: Rockwell Mfg. Co., Hydraulics Div., 50 Water St., Pittsburgh 22.

For additional information circle MD 26 on Page 271

Finish for Aluminum

Protective finish for aluminum is corrosion resistant and has good paint-anchoring properties. The coating, tradenamed Alodine, is of the chemical type and is applied by dipping, spraying or brushing. Time required for application varies from about 20 seconds to 5 minutes depending upon method of application. Manufacturer: American Chemical Paint Co., Ambler, Pa.

For additional information circle MD 27 on Page 271

Magnetic Starters

Bulletin 4111, Size 1, ac magnetic starters are designed for nonreversing, across-the-line starting of polyphase, squirrel-cage, induction motors as well as single-phase motors. They are suitable for use with separate pilot devices for 2 or 3-wire remote control or can be supplied with local control pushbutton stations or selector switches. Units are designed in accordance with NEMA and Underwriters Laboratories standards for industrial control and are available with open construction or NEMA Type 1 general-purpose enclosures. Features include high arc-interruption capacity, fast operation, thermal overload protection and complete accessibility of parts. Maximum enclosed rating is 7½ hp at 440-550 volts, 3-phase. Standard operating coils can be furnished for 110, 210, 440 and 550 volt service at 25, 50 or 60 cycles. Dual voltage coils can be supplied for 110-220 or 220-440 volt, 60 cycle service. Manufacturer: Ward Leonard Electric Co., Mount Vernon, N. Y.

For additional information circle MD 28 on Page 271

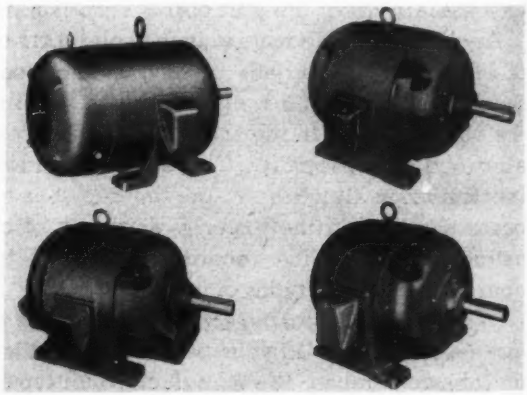
Synthetic Enamel

Line of synthetic enamels tradenamed Lustral 67 is recommended for general industrial painting and finishing. It is suitable for brushing, spraying or dipping and may be air dried or baked. It is intended primarily for interior use although it may be used satisfactorily for outside service. Color range of the new line includes 21 colors in addition to white and black. Manufacturer: Sherwin-Williams Co., industrial division, 101 Prospect Ave., Cleveland 1.

For additional information circle MD 29 on Page 271

Polyphase Motors

Manufactured in integral horsepower ratings in all commercial frequencies, new line of polyphase industrial motors is available with a variety of enclosures. These include drip proof, splash proof, totally en-



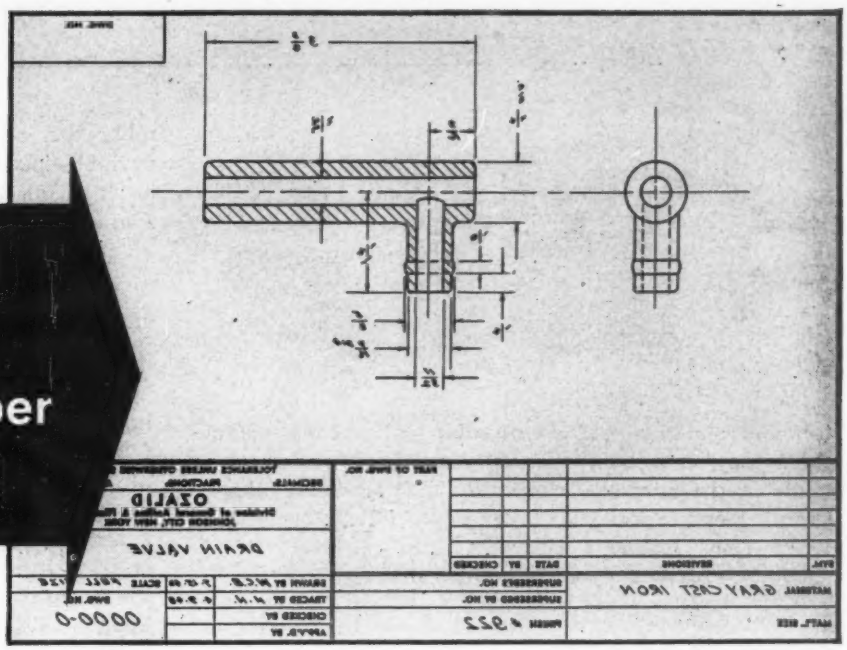
closed fan cooled, and totally enclosed nonventilated. Manufacturer: Electric Boat Co., Electro-Dynamic Div., Bayonne, N. J.

For additional information circle MD 30 on Page 271

3-Way Air Valves

Poppet-type 3-way air valves have 5/16-inch seat openings allowing the high air flow necessary for quick acting of large air cylinders. The units are fitted with hose nipples to take hose ranging from ⅜-inch ID to ¾-inch OD. Mounting holes at both

You're looking at the NEW intermediate paper for mass users!



The drawing above was reproduced on the new Ozalith Intermediate Paper. This duplicate paper original gives you Ozalid prints of maximum line density.

This New Ozalith Paper Costs Little—Has Great Mechanical Strength—Washable, Plastic Surface!

If your operation demands the duplication of a large bulk of drawings and plans, this new Ozalith Intermediate Paper should be of great interest to you.

For it is the answer to the need for a reasonably priced (7½¢ per square foot) paper duplicate original having maximum possible toughness of base; permanence, for filing and record use; and highest reprint quality. It reprints at exceptionally high machine speed.

No Tendency To "Bleed"

The dye image will not offset or transfer to other papers or tracings with which this new Ozalith paper has been placed in contact. There is no tendency to "bleed."

Ozalith is coated on a 100% rag base of great mechanical strength. It will not deteriorate appreciably with age. It is the most durable intermediate paper known.

Pen And Pencil Additions

You can wash Ozalith with a damp cloth—dust, dirt, and grease will come off quickly. Water will not cause the base to cockle or the image to run.

Pen or pencil additions can easily be made on either side of an Ozalith print. Because of its high transparency, contact prints should be made, and additions then made on the unsensitized surface.

Ozalith is a running mate to Ozacloth—but because of its low price, it is ideal for mass users. Write, today, if you would learn more about this and other Ozalid prints.

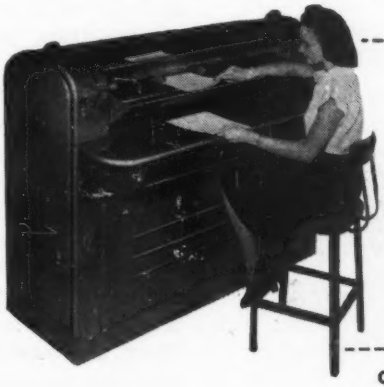
ALL OZALID PRINTS PRODUCED IN SAME MANNER

★ **No tie-ups when you shift from one type of print production to another. Simply choose your Ozalid material . . . and your Ozalid print-making machine exposes and dry develops it. Standard work prints are produced in 25 seconds.**

★ **Your drawings can be up to 42 inches wide, any length. Roll stock or cut sheets can be used. (Special machines accommodate 54" wide drawings.)**

★ **You—or anyone else—can be the operator. A few hours and you're an "expert."**

★ **See *all* the Ozalid prints you may make from any drawing . . . and learn full story. Mail coupon today.**



DEPT. NO. 215

OZALID . . . A Division of General Aniline & Film Corp., Johnson City, New York

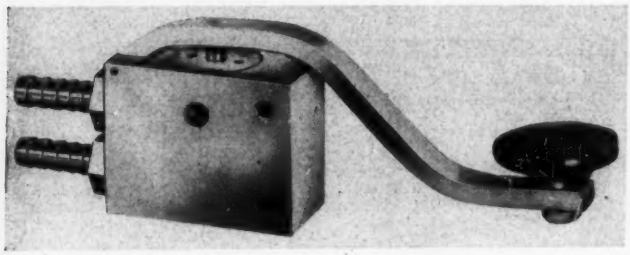
Gentlemen: Please send free copy of Ozalid Streamliner booklet illustrating all types of Ozalid prints.

Name_____ Position_____

Company_____

Address_____

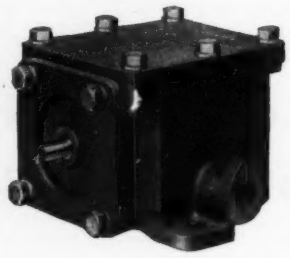
Ozalid in Canada—Hughes Owens Co., Ltd., Montreal



top and sides of the rectangular valve body make them easy to attach. Presently available models are made in the normally closed or "off" type. Manufacturer: Mead Specialties Co., Dept. FT-25, 4114 N. Knox Ave., Chicago 41.

For additional information circle MD 31 on Page 271

Centrifugal Switch

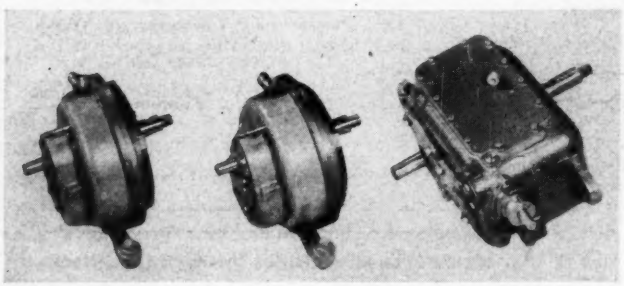


Centrifugal switch is designed to meet Underwriters' Laboratories specifications for explosionproof equipment to be used in hazardous atmospheres of Class I, Group D and Class II. The units are designed for use as overspeed switches, underspeed switches, zero-speed plugging switches, and conveyor interlocking switches. In this last application centrifugal switches with normally open contacts are installed near the junction points in belt conveyors. In case of stoppage of the main belt, the electrical contact in the switch opens and automatically cuts off power to the feeder belt. Manufacturer: Euclid Electric & Mfg. Co., Madison, Ohio.

For additional information circle MD 32 on Page 271

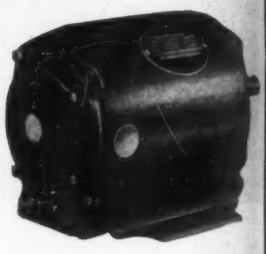
Gear Transmissions

Line of gear transmissions is made in the horsepower range 0.1 to 7 and in reduction ratios of 40:1, 17:1 and 4:1. Several units in the line are two-speed types with gear shifts for convenient speed change. Each transmission is a self-contained unit with integral means for mounting. Ball and roller bearings and hardened gears are used throughout; lubrication is provided by means of an oil bath. The units are recommended for use with Speedmaster variable-speed transmissions to provide speed ratios as high as 30:1. Manufacturer: DoAll Co., Des Plaines, Ill.



For additional information circle MD 33 on Page 271

Fractional-Horsepower Motors



Series of fractional horsepower motors known as the B-Line features the use of built-in resilient mountings. Consisting of large square-rubber cushions, encased in steel cradles, the mounts are located inside the motor end covers. They are, therefore, protected against mechanical abuse and deterioration. Additional advantage is the improved appearance and the absence of dust-catching surfaces. Units are made in ratings from 1/6 to ¾-hp and are recommended for such applications as heaters, refrigerators and blowers. Manufacturer: The Brown-Brockmeyer Co., Box 976, Dayton, O.

For additional information circle MD 34 on Page 271

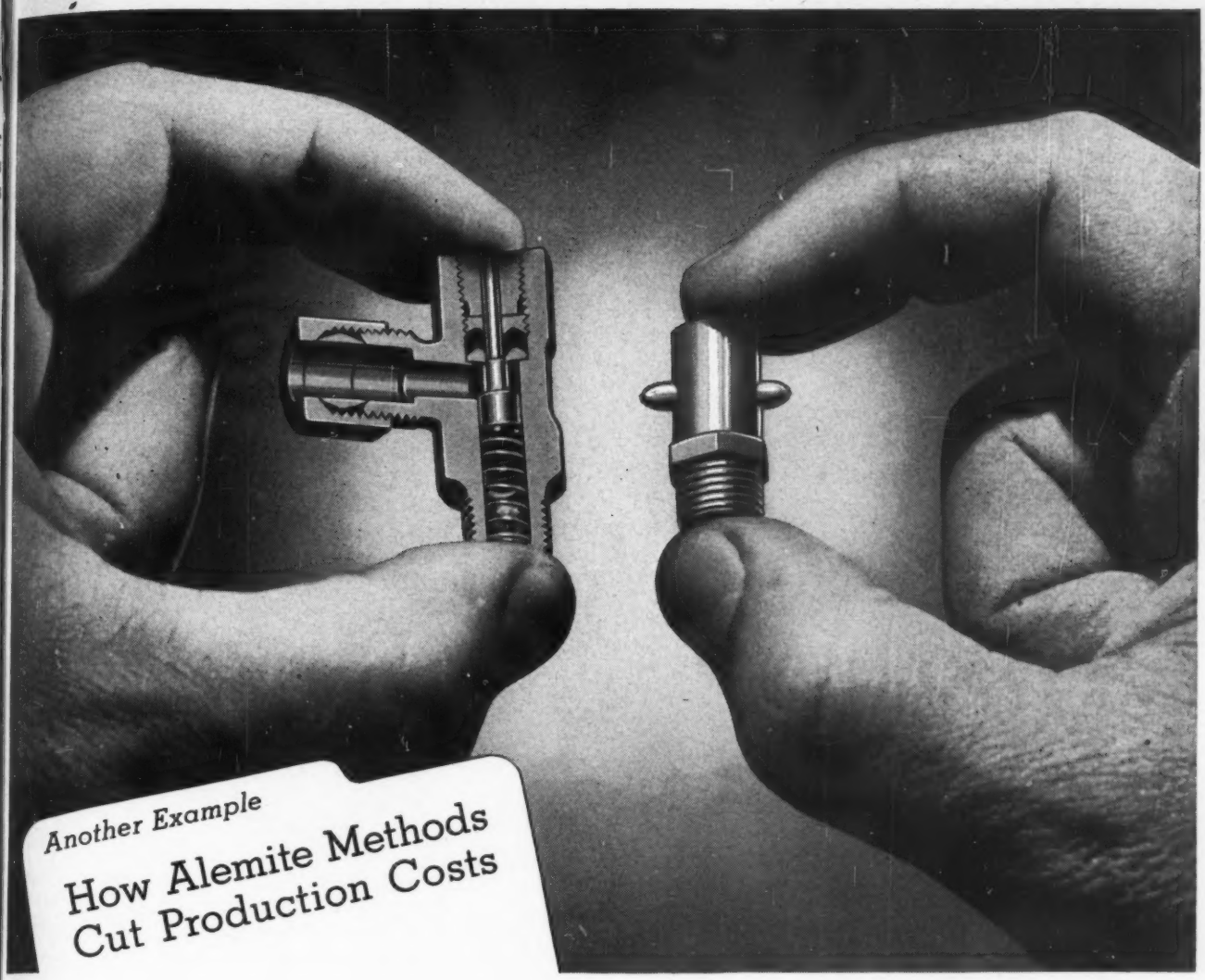
Spiral Retaining Rings

Retaining ring known as the Spirolox is made of a flat steel band wrapped in a double helix. It is thus concentric in its groove at all times and gapless. Because of this the ring is dynamically balanced when installed and has equal holding strength at all points around its circumference. It is said to have resistance to centrifugal force and resist winding or unwinding under thrust force. Installation in inside or outside grooves requires no special tools. Manufacturer: Ramsey Corp., St. Louis, Mo.

For additional information circle MD 35 on Page 271

Combination Starters and Circuit Breakers

Requirements for motor branch circuit overcurrent protection and disconnect means is provided by Bulletin 4130 alternating-current combination magnetic starters and circuit breakers. Each unit consists of a magnetically operated contactor with thermal inverse-time delay overload relays, combined with a thermal type circuit breaker. The components are housed in an NEMA Type I enclosure. Operated from outside the enclosures the circuit breakers have provision for padlocking in the off position. They are interlocked mechanically to prevent opening enclosure door when the breaker is on. These



"Midget" Centralized Lubrication System Helps Machines Produce More

A LOW COST SYSTEM THAT'S HARDLY LARGER THAN AN ALEMITE PIN-TYPE FITTING

Protect Those Machines You Design from Human Error

Human error in lubrication can easily destroy the efficiency of the best designed machine. A five minute conference with an Alemite Representative may well reveal lubrication design features that will make your machines produce more. This man is fully qualified to help you because he is a specialist in modern methods of handling and applying lubricants. Write to Alemite, 1804 Diversey Parkway, Chicago 14, Illinois.

Small, Simple, Compact . . . this new Alemite "Midget" measuring lubrication valve is designed for all types of heavy, light and precision machines—especially where space limitations are encountered. With this single line system, valves mounted on or near the bearings deliver a measured amount of lubricant from one safe, central point while machines keep right on producing!

Alemite Centralized Lubrication Systems make possible the lubrication of hundreds of bearings from one central point—greatly reduce the possibility of "human error". There is no danger of over-lubrication or under-lubrication . . . handles grease or oil. Oilers work in complete safety. Result: Production goes up . . . costs go down.

Just one of many ways that Alemite Methods help industry add productive time to machines with faster, more efficient lubrication equipment and procedures. The Alemite Representative can show case after case where a time study analysis has proved that his methods reduce costs. These methods eliminate costly, time-consuming handling of oils and greases . . . slash "time-out" for lubrication . . . keep dirt and moisture out of lubricants . . . completely mechanize lubrication from barrel-to-bearing. These benefits all add up to lower production costs and longer machinery life. Get the complete facts, today.

ALEMITE

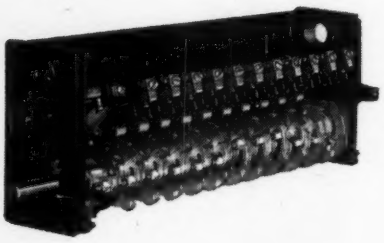
Another Product of STEWART WARNER

MODERN LUBRICATION METHODS THAT CUT PRODUCTION COSTS

controls are available in 5 standard sizes ranging up to 200 hp, 3 phase, 550 volts, 60 cycles, maximum. They can be supplied with only reset button in the cover, with local control start-stop pushbuttons, or "Hand-Off-Auto" selector switches. Manufacturer: Ward Leonard Electric Co., Mount Vernon, N. Y.

For additional information circle MD 36 on Page 271

Rotating-Cam Limit Switch



Up to 12 circuits can be opened and closed using the Bulletin 801 heavy-duty rotating cam limit switch. Circuits can be opened or closed in any sequence. Each cam is indexed for accurate setting, contactors are individually detachable and the shaft and cam assembly is easily removed for adjustment. Units are also available for 3, 4, 6 or 9 circuits. The switches are adaptable to most programming or time-sequence operation and are available in general purpose or watertight enclosures. Manufacturer: Allen-Bradley Co., Milwaukee.

For additional information circle MD 37 on Page 271

Diesel Engine

Diesel unit, designated the HR-600 engine, develops 165 hp maximum at 1800 rpm. It has a 5⅛-inch bore and a 6-inch stroke. The new diesel is made in three styles: An automotive model is designated type HRB-600; three industrial types are HRB1-600, HRI-600 and HRP-600; and a marine model is type HRM-600. Of the three industrial units, the first is the basic model; the second is equipped with instrument panel and trunnion support base; while the third is an enclosed type with structural-steel base, clutch power takeoff, radiator, fan, engine hood and fuel tank. Manufacturer: Cummins Engine Co. Inc., Columbus, Ind.

For additional information circle MD 38 on Page 271

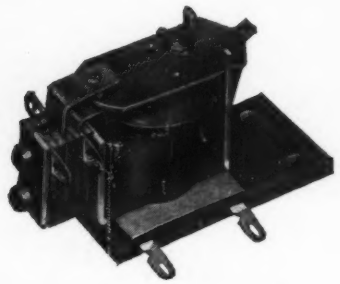
Metal-Cased Mercury Switches

New mercury switches use hydrogen gas as arc suppressor, reducing electrode corrosion and providing long life. Electric-welded steel case seals the gas under pressure, keeping pressure intact whether switch is in use or inactive. There is said to be little deterioration over many months. Metal case of the switch acts as one contact. Units are especially efficient in high-capacity, highly-inductive circuits or where variable loads place undue strain on switches. Models range from 1 to 65 amperes in capacity and are made with or without plastic protective covering. Rubber insulators may be had to provide waterproofing. Manufacturer: Durakool Inc., Elkhart, Indiana.

For additional information circle MD 39 on Page 271

Sensitive DC Relay

Type E single-pole, double-throw relay is a sensitive dc unit designed for use with electron-tube circuits and similar applications. Measuring 2½ by 1¼ by 1½ inches, the relay is recommended for use where space is a consideration. It has a contact capacity



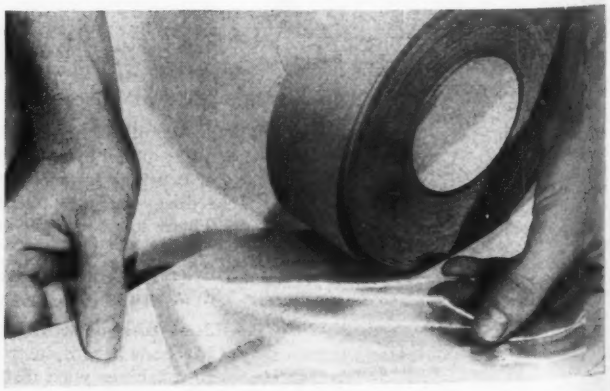
of 5 amperes at 25 volts noninductive load and power consumption is 60 milliwatts. Manufacturer: Comar Electric Co., 3148 N. Washtenaw Ave., Chicago 18.

For additional information circle MD 40 on Page 271

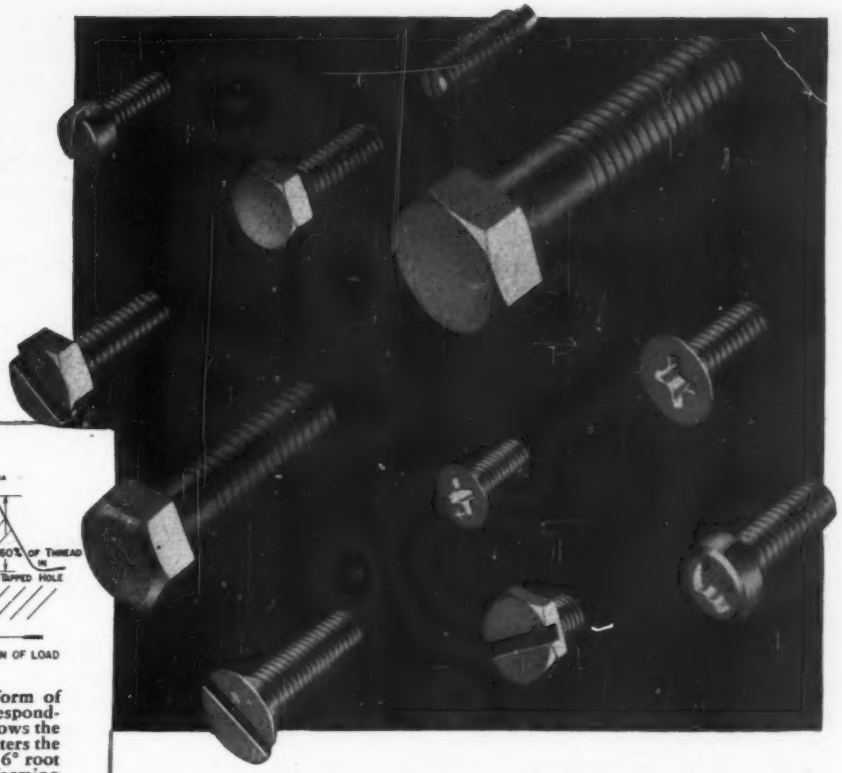
Bonding Film

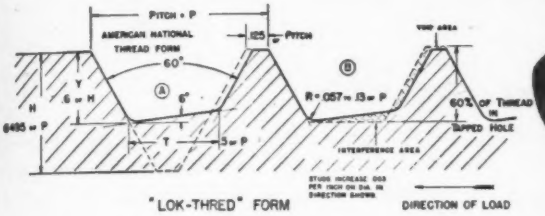
Bonds with shear strengths up to 3500 psi can be made using new sheet-type adhesive. The material, known as Scotch-Weld bonding film has no filler material. It can be used to join parts of many materials by the application of pressures from 25 to 100 psi, and temperatures from 300 to 500 F. Manufacturer: Minnesota Mining and Mfg. Co., 900 Fauquier Ave., St. Paul, Minn.

For additional information circle MD 41 on Page 271



the only thread that OF ITSELF holds tight





The heavy black line at position A shows the form of "Lok-Thred", while the dotted line shows the corresponding American National Thread form. Position B shows the action which occurs when the male "Lok-Thred" enters the 60% American National Thread tapped hole. The 6° root angle, through the interference created, causes a reforming of the socket thread, which fills the void area and causes a locking action, plus a positive seal.

IN "LOK-THRED" NATIONAL SCREW OFFERS AN IMPORTANT NEW TYPE OF BOLTS AND SCREWS

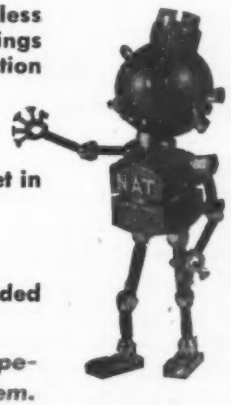
"Lok-Thred" fasteners actually become tighter in service, even under vibration. They are available now in bolts and screws as well as studs, and with special or standard heads.

They not only lock more securely than American National Threads (see diagram), but also seal positively, even against liquids under pressure. Bosses and blind tapping can be eliminated.

National PRODUCTS

Note These Important "Lok-Thred" Advantages

1. Lock securely and become tighter in service.
2. Have much higher fatigue limits than fasteners with conventional threads.
3. Stronger in both tension and torsion than ordinary American National Threads.
4. Carry entire normal working load on 6° angle at root of thread under high compressive prestress.
5. Modified American National Threads permit use of standard tools.
6. Re-usable and on any re-application less than one-half additional turn brings torque back to its original installation value.
7. Do not require selective fits.
8. Do not gall when being driven nor fret in service.
9. Act as dowels and taper pins.
10. Seal positively and eliminate added bosses and blind tapping.



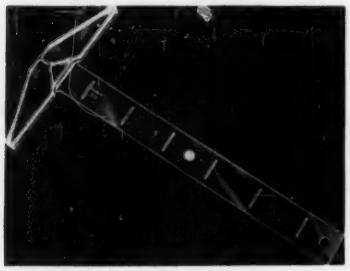
Write for "Lok-Thred" booklet, or send specific information on your fastening problem.

THE NATIONAL SCREW & MFG. COMPANY, CLEVELAND 4, OHIO

engineering dept equipment

In order to obtain additional information on this new equipment see Page 271

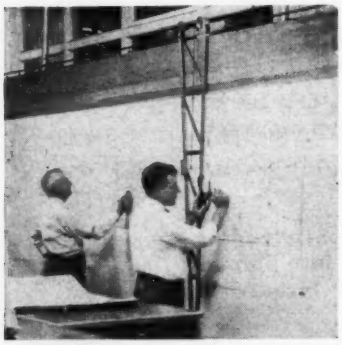
Molded Plastic T-Square



Molded in one piece of strong plastic, new T-Square is said to remain accurate even if dropped. This drafting tool is made of a transparent plastic which permits full visibility of the surface over which it is placed. Both sides of the instrument may be used because of the double-head design. Ribs, which protrude 0.020-in. beyond both surfaces of the shaft, allow the T-square to be moved across the working surface smoothly; in addition, the ribs raise the square above the paper to facilitate inking. The instrument is available in two blade lengths; 18 and 24 inches. Manufacturer: Instrumaster Industries, 2457-62 W. Jackson Blvd., Chicago 12.

For additional information circle MD 42 on Page 271

Stainless-Steel Drafting Machines



All-metal drafting machines are suitable for use with horizontal, vertical or inclined boards. Of stainless - steel construction the machines incorporate a micrometer quadrant giving rapid selection of angles at 3-degree intervals with fine adjustment to 2½ minutes. Rapid indexing is provided by disengaging the adjusting screw making it possible to slip the straightedge around the quadrant rapidly to the desired position. Counterbalancing is provided by spiral spring and motion in a horizontal plane is provided by a ball-bearing track located at the top of the board. Support over the drawing is provided by plastic rollers which reduce smudging to a minimum. A variety of interchangeable scales is available for use with the machine. Models of these drafting machines are made to fit boards with heights of 24, 30, 36, 48, 54, 60, 72, 84, 96, 108, 120, and 132 inches. They will fit any length board. Special machines including left-handed types, can be furnished for odd-sized drawing boards. Manufacturer: Emmert Mfg. Co., Waynesboro, Pa.

For additional information circle MD 43 on Page 271

Drafting-Instrument Set

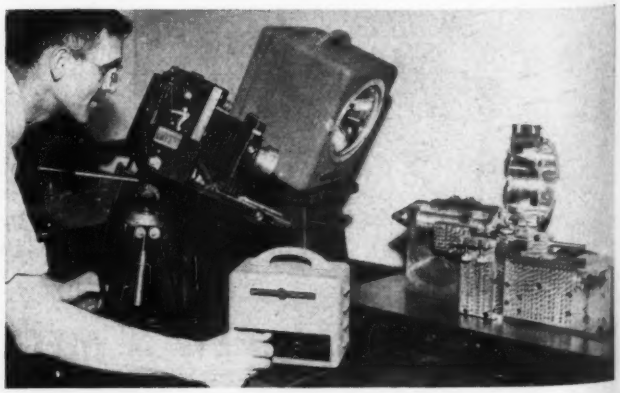
Model A-171-V drafting instrument set consists of a 6½-inch bow compass, a 4½-inch bow compass, a friction type divider and a ruling pen. The larger compass will describe circles up to 10½ inches in diameter while the smaller will describe circles from 1/16 to 6 inches in diameter. The divider opens to 7 inches. A pen attachment can be used with either compass to convert them into pen compasses. The instruments are packed in a leatherette velvet-lined case, with a cowhide case also available. Manufacturer: V. & E. Mfg. Co., 758 S. Fair Oaks Ave., Pasadena 20, Calif.

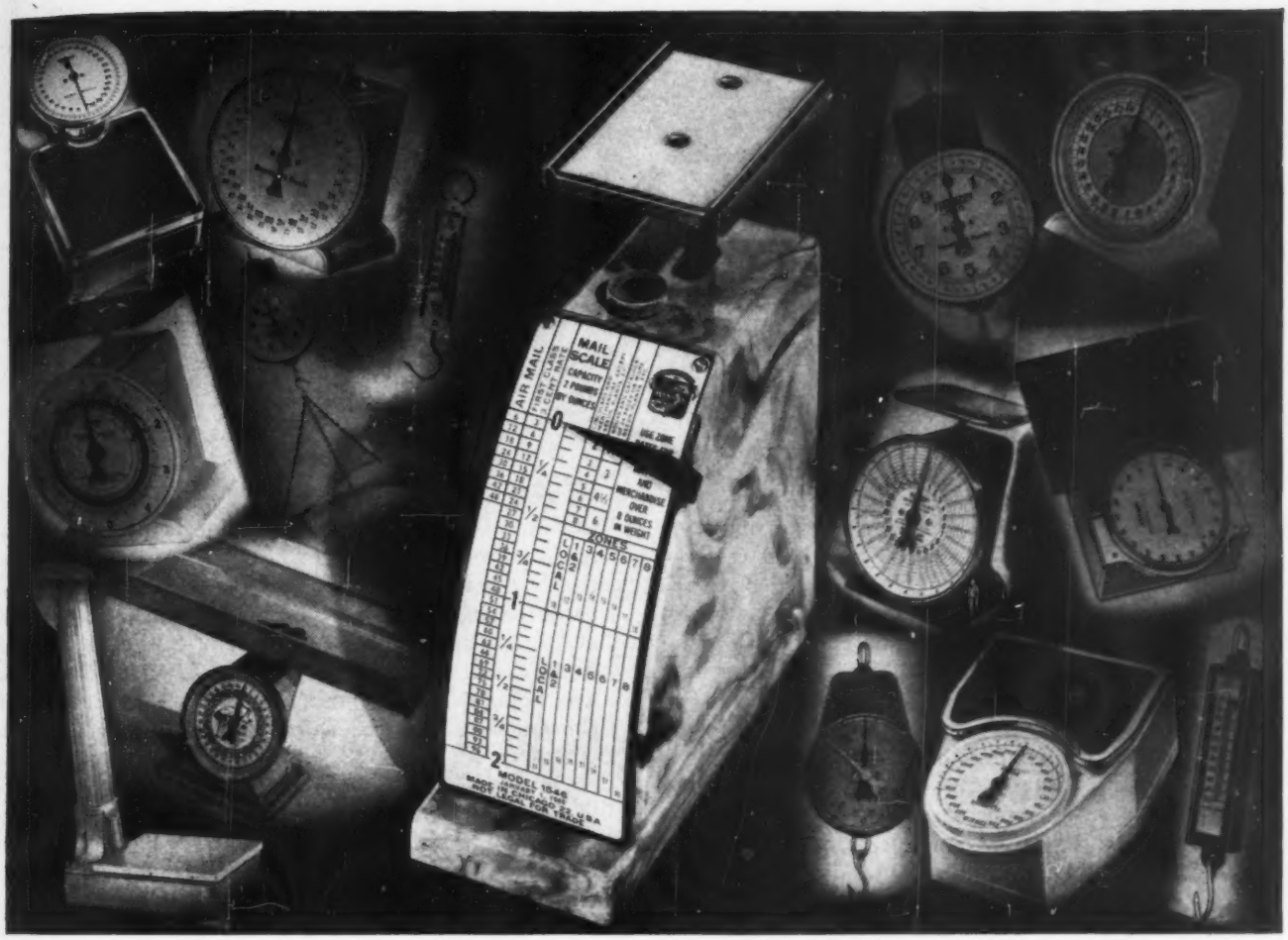
For additional information circle MD 44 on Page 271

High-Intensity Photolight

High-intensity blue-white flash of two microseconds duration is provided by new photolight, useful in photographer analyses of vibration, as well as high-speed rotational and linear motion, etc. The light unit weighs 25 lb and consists of a three-electrode, short-gap, high-pressure, gas-filled tube together with a fixed-focus reflector and a capacitor-discharge type power supply. It operates from a 115-volt single-phase power supply. Flash can be initiated by closing normally-open contacts, opening normally-closed contacts, by a pulse from a crystal microphone or by a phototube pickup. Manufacturer: General Electric Co., Schenectady 5, N. Y.

For additional information circle MD 45 on Page 271





TYPE "A" ASSEMBLY BIT

Again... CLUTCH HEAD *Screws Scale Down Assembly Costs...*

ELIMINATE SKID DAMAGE AND SMOOTH OUT PRODUCTION

HANSON, a Chicago name nationally recognized as symbolic of weight accuracy, has this to say as to why their complete line of 1949 models is being standardized on CLUTCH HEAD Screws:—

"Compared with the use of other recessed head screws, change-over to CLUTCH HEAD has given us immediate relief from several costly and annoying assembly problems."

"Over and above faster, smoother operation, the non-tapered drive has freed us from the expense and delay arising out of skid damage."

"The Lock-On has definitely simplified for us screw placement in 'awkward' spots in our assembly."

"The durability of the Type 'A' Bit, plus its easy reconditioning, is a time and money saver. Our tool maintenance cost is now almost zero."

"The fact that CLUTCH HEAD operates with a common screwdriver rates high with us as a service factor."

COMMON SCREWDRIVER

Regardless of the size or type of screw required, these and other exclusive CLUTCH HEAD *advantages will prove their superiority and economy in every phase of assembly production. Their value will be immediately apparent on cursory examination, so we invite you to send for screw assortment, sample Type "A" Bit, and illustrated Brochure . . . to come to you by mail.*

CLUTCH HEAD · SCREWS ·

UNITED SCREW AND BOLT CORPORATION

CLEVELAND 2 · CHICAGO 8 · NEW YORK 7

MEN...of machines

JAMES M. TODD, consulting engineer of New Orleans, is the newly elected president of the American Society of Mechanical Engineers. Mr. Todd received his degree in mechanical engineering from Tulane University in 1914. After service in the engineer corps in World War I he served as chief engineer of the Marrero plant of Penick and Ford, and later with A. M. Lockett and Co. In 1928 he established a private practice as consulting engineer in the mechanical and electrical engineering fields, specializing in air conditioning, power plants and factory equipment. He is recognized as being an authority in the mining and production of crude sulphur, having been vice president in charge of engineering and member of the board of the Jefferson Lake Sulphur Co. During World War II Mr. Todd handled mechanical and electrical work in connection with airfields, shipyards and other defense projects. He was president of the Louisiana Engineering Society in 1930 and served for six years on its board. The new ASME president has been active in the national and local affairs of both that society and the American Institute of Electrical Engineers. He organized the New Orleans section of the AIEE and was its first chairman. Mr. Todd was chairman of the New Orleans section of the ASME for three years, has been active on several national committees, and has been society vice president.

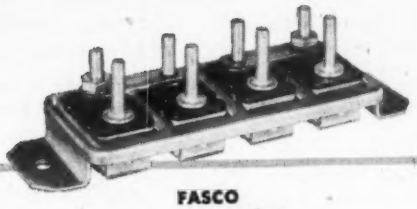
PAUL E. BUTZIN has been appointed director of engineering of the Delta Manufacturing division of the Rockwell Manufacturing Co., Milwaukee. He was the first permanent design engineer hired by the company, having been given his first position at Delta in 1933 after serving as supervisor of tool design of the A. O. Smith Corp. for six years. In 1942 Mr. Butzin was appointed chief engineer of the Milwaukee Gear Co. and later became works manager. He has been active as a product and machine designer in the machine tool, electrical, and steel fabricating industries since 1916. Mr. Butzin is a member of the American Gear Manufacturers Association and the American Ordnance Association. He is president of the Industrial Club of the Employers Association of Milwaukee, and is a past chairman of the Milwaukee chapter of the American Society of Tool Engineers.

James M. Todd

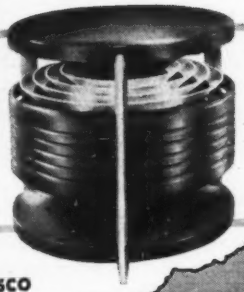
Paul E. Butzin

Fred B. MacLaren

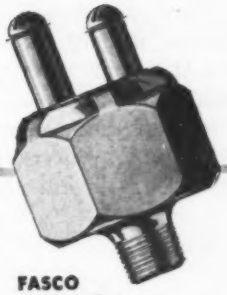
FRED B. MACLAREN has joined the Servo Corporation of America as head of the servomechanisms laboratory. Mr. MacLaren graduated from Yale University in 1936 and became associated with programs for railroad electrification and the development of industrial controlling and recording equipment. During the war he was in charge of antiaircraft fire control development at the Frankford Arsenal. Since 1946 Mr. MacLaren has had extensive experience in the fields of electronic control mechanisms and com-



FASCO
Automotive
Circuit Breakers



FASCO
Table Type Fans



FASCO
Automotive
Stop Light Switches

~~F. A. SMITH MANUFACTURING CO., INC.~~
ROCHESTER 2, NEW YORK

Announcing

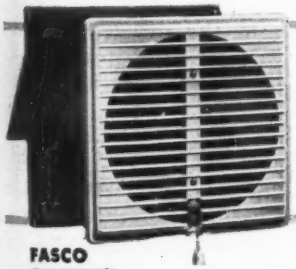
A NEW COMPANY NAME

FASCO INDUSTRIES, INC.

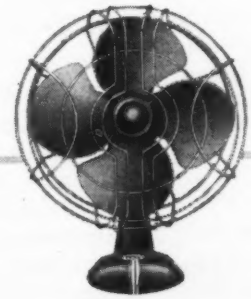
Our corporate name now conforms more closely with the trade-mark, "FASCO," used on our products. FASCO represents the *same company* . . . carries on our same long-established reputation for *dependable quality* . . . *superior performance* . . . *right prices*. From now on you can identify our products and company by the names "FASCO" and "FASCO INDUSTRIES, INC."

H. P. Toppin, PRESIDENT

H. M. Sylvester, SECRETARY-TREASURER



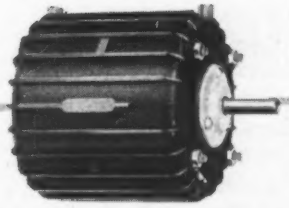
FASCO
Automatic
Kitchen Ventilators



FASCO Desk Fans

FASCO
Blowers

FASCO Fractional H. P. Motors



FASCO Fractional H. P. Motors

puters for fire control systems. He is in charge of the company's project activities in the servomechanism field.

♦

GEORGE N. SIEGER was recently elected president of the American Welding Society for the coming year. Mr. Sieger, president of the S-M-S Corp., was installed as head of the society at the annual meeting during the week of October 24. He is a graduate of Lehigh University. A national authority on resistance welding, he is a past president of the Resistance Welder Manufacturers Association. During the war he served on several committees of the War Production Board, and was a consultant on cemented tungsten carbides for the cutting tools section of the board. Mr. Sieger has held many offices in the AWS, including the first vice presidency in 1948. He is a member of the American Society for Metals and the Society of Automotive Engineers.

♦

JAMES R. LONGWELL, for the past four years director of engineering and research, is now assistant to the president of the Carboloy Co. Inc., Detroit. Mr. F. C. RITNER, with the Carboloy company since 1929, has been elevated to the position of vice president in charge of engineering and research.

♦

PAUL BRAINARD has been named head of the engineering standards department recently organized by the Hyster Co., Portland, Ore. Under Mr. Brainard's direction this department will work toward the standardization of design and materials in the company's three plants located in Portland, Ore., Peoria and Danville, Ill. Previously Mr. Brainard was assistant supervisor of hoist design.

♦

G. WAINE THOMAS, formerly assistant chief engineer of the transportation-agricultural-industrial division of Continental Motors Corp., has been appointed chief engineer to succeed EARL C. GINN who is now executive vice president and director of engineering.

♦

T. J. ZELLER has been promoted to factory manager of the Allentown, Pa. plant of Mack Trucks Inc. In assuming direction of the company's largest plant, Mr. Zeller replaces Mr. C. J. MORAN who has been transferred to Mack's western sales headquarters. Mr. A. C. SCHLIEWEN has been appointed to the post vacated by Mr. Zeller, that of manager of the Plainfield, N. J. plant.

♦

C. W. STAACKE, one of the country's leading belt conveyor engineers and a member of the B. F. Goodrich Co. engineering staff for 22 years, has joined Hewitt-Robins Inc. as a technical consultant. Mr. Staacke, who will be special assistant to the president, is credited with several major improvements in conveyor belt design, including the introduction of cord fabric in conveyor belt carcass. His duties will include serving as technical co-ordinator of the company's research and development activities. Mr. Staacke received his mechanical engineering degree from Ohio State University, and during the war served as a special consultant to the air forces of the Army and Navy on fuel cell development.

♦

BYRON H. SHINN, formerly with Firestone Aircraft Co., Willow Grove, Pa., has become research designer for Pratt & Whitney Aircraft Co., East Hartford, Conn.

♦

HOWARD A. REED was transferred recently to the Avon Lake, O., plant of the Fruehauf Trailer Co. as resident engineer in charge of the engineering department. He previously was associated with the chassis division of the company's engineering department in Detroit.

♦

HARRY CONN has recently accepted the position of chief engineer in charge of the engineering division of Scully-Jones & Co., Chicago. In addition to being an experienced designer, Mr. Conn has a background as an executive and consultant in the mechanical field. For several years he was associated with the International Harvester Co., and in 1942 he set up and directed the LaSalle Engineering Co. in New York. He also has served as chief engineer for the Gridley Farm Tools Co. Mr. Conn is a member of the American Society of Tool Engineers and the American Society for Metals.

♦

JOEL D. JUSTIN, consulting engineer of Philadelphia, has been elected chairman of the Engineering Foundation. Dr. Boris A. Bakhmeteff was chosen vice chairman.

♦

CARL J. LAMB has been appointed consulting engineer of the Hydraulic Press Mfg. Co., Mt. Gilead, Ohio.

♦

LOREN F. JONES, manager of research and development projects, Radio Corp. of America, and HUGH H. SPENCER, manager of Teleran sales, recently were awarded Presidential certificates of merit in recognition of their contributions in the development of direction-finding equipment and guided missles during World War II.

♦

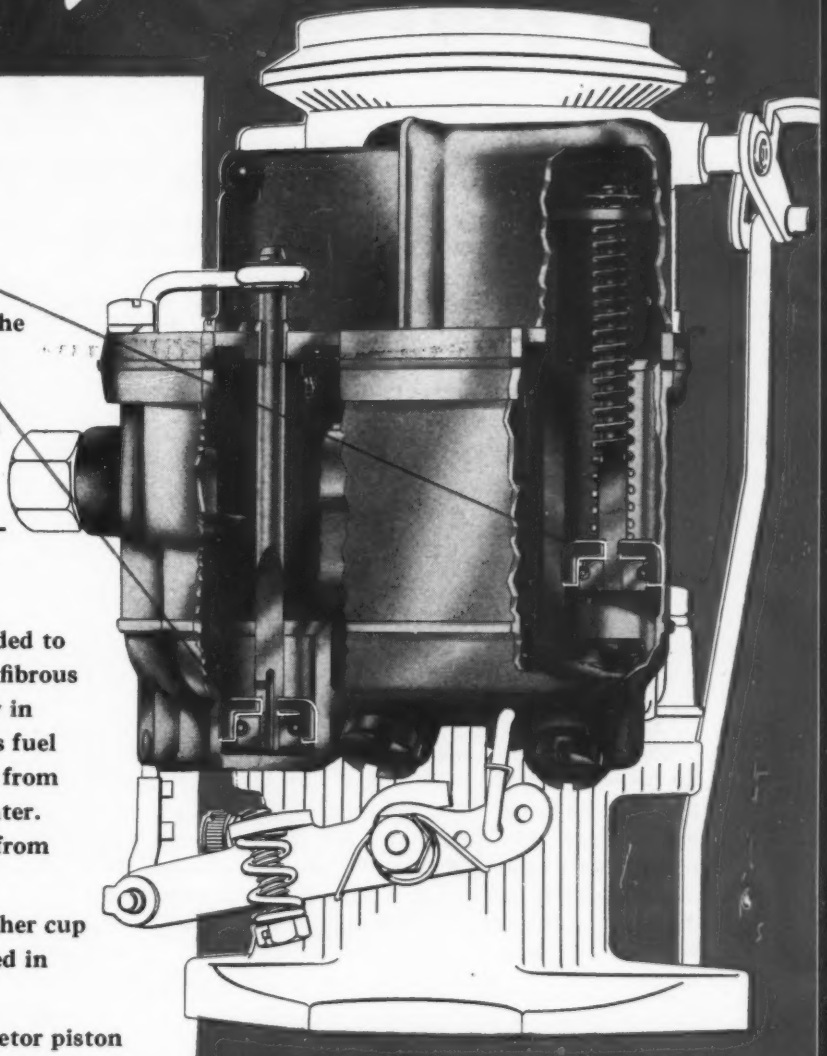
W. E. WALTERS upon retiring as president of the British Institution of Engineering Draftsmen and Designers, was unanimously named a fellow of the institute in recognition of his original analysis of three-dimensional projection as applied to engineering drawing. Articles on this subject by Mr. Walters have been published in MACHINE DESIGN.

♦

DONALD B. OLEN has been named director of the engineering division, Four Wheel Drive Auto Co. C. D. SIMONDS has been appointed chief engineer in charge of design of the company's standard models and B. G. DONLEY becomes manager of development engineering.

• IN ENGINEERING, AS IN DRIVING

it's little things that count



Like that little squirt of gasoline, when you want quick starting and fast pick-up. And the little piston pump packing which helps you get it. In Stromberg carburetors (made by Bendix Aviation Corp.), special care is taken to assure complete dependability of *every* component part. In model BXVD-3, for instance, two leather cup packings are used—one on the dash pot piston and the other on the pump piston.

These cup packings must be accurately molded to size; they must *stay* wrinkle-free; and their fibrous texture must have that tightness found only in prime calfskin. These cups must not by-pass fuel or become logged; they must neither shrink from heat nor freeze to their cylinder walls in winter. Their leather tannage must not deteriorate from contact with oils or gasoline.

Non-deteriorating *chrome* tanned Sirvis leather cup packings, made by Chicago Rawhide, are used in thousands of Stromberg carburetors.

Chicago Rawhide engineered the first carburetor piston pump packing to deliver satisfactory performance. Today, they are used in many leading makes. Because of constant research and product development, precise laboratory control, highest standards of leather quality, and exceptional care in every phase of production, Chicago Rawhide's Sirvis leather piston pump packings continue to be the most dependable.

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1304 Elston Avenue Chicago 22, Illinois

for assured performance

in your leather applications—whether in connection with oils, grease, water, or air, under high, low or static pressures—specify SIRVIS. Your inquiries will be promptly answered.

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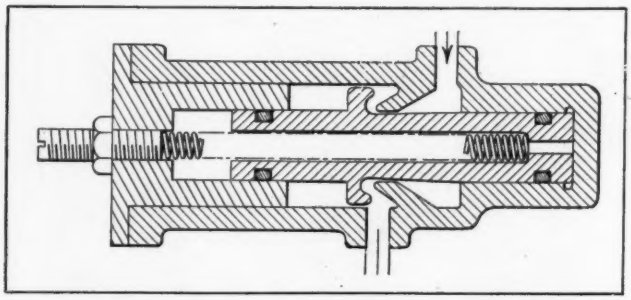
MECHANICAL LEATHERS

OTHER C/R PRODUCTS
PERFECT Oil Seals
SIRVENE Synthetic Rubber

Noteworthy Patents

CONSTANT FLOW of hydraulic fluid regardless of variations in input or back pressures is provided by a flow-control valve described in patent 2,441,220 assigned to United Shoe Machinery Corp. by Paul H.

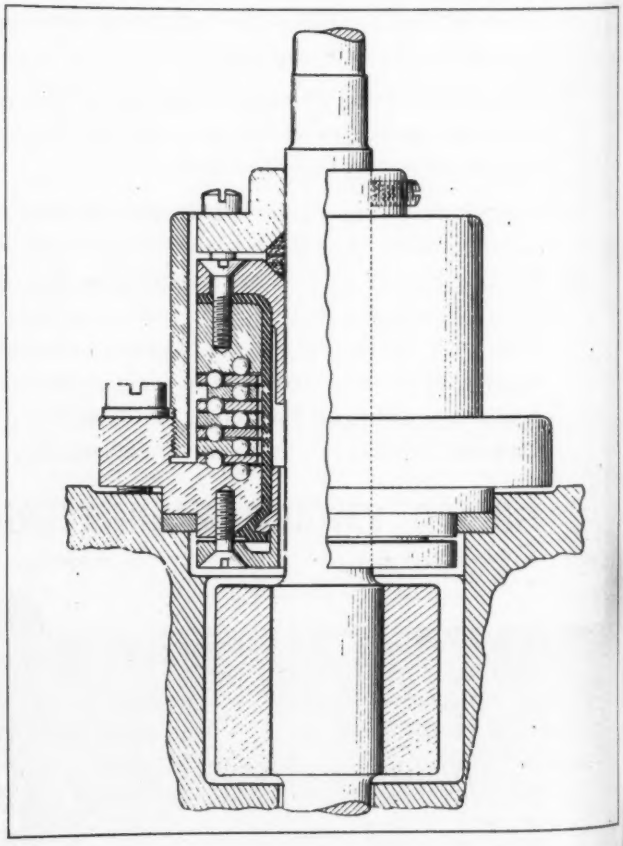


Dixon. The valve has an inward flange portion in the valve body which creates a variable restriction in the fluid passage. The jet formed by the fluid passing between this flange and the spring-loaded valve plunger acts upon a flange on the plunger to position the plunger in accordance with the fluid velocity to maintain substantially constant flow through the valve at all times.

EXTREMELY LONG hydraulic cylinders, ordinarily difficult to cast, handle and finish machine, can be readily produced by means of a new method of design described in patent 2,443,312. Assigned to H-P-M Development Corp. by G. M. Geiger and H. A. Donnewirth, the method utilizes a sectional arrangement to provide long cylinders and overcome production difficulties. Separate sections are screwed together with an O-ring packing and after assembly of the desired numbered of sections, the internal bore is ground or honed to the required accuracy.

MINIMUM ROTOR INERTIA and small size are attained with a polyphase servomotor design patented by John P. Glass. Assigned to the Kerfott Co. under patent 2,443,104, the motor is designed to use preformed exciting coils which are disposed in the housing instead of on the individual pole pieces. The coils carrying the two phases of the exciting voltage are arranged to avoid inter coupling so that when one phase is de-energized, no torque is produced. This feature is important inasmuch as follow-up systems use one constantly energized phase and, if coupling is present, the second-phase timed winding to which the control signal is sent often assumes shading pole characteristics causing undesirable drift.

EFFECTIVE SEALING at high fluid pressures is made possible with a new seal design intended for use on shafts of sensitive indicating instruments which are required to turn only a fraction of a revolution. Covered in patent 2,441,618 assigned to Pitometer Log Corp. by Edward S. Cole, the unit utilizes a tubular sealing element of rubber which is connected to the stationary housing at one end and to the shaft at the other. Under high pressure such a seal ordinarily would burst when twisted and would



General Electric Announces

A NEW LOW-COST THY-MO-TROL DRIVE

TYPE H1 in sizes up to 1/2 horsepower

. . . adjustable speed from a-c power

Provides the same easy adjustment features and wide speed-range characteristics of the larger Thy-mo-trol drives, but it utilizes a simplified half-wave circuit. Available in sizes up to ½ horsepower.

FAST, ACCURATE PRODUCTION . . .

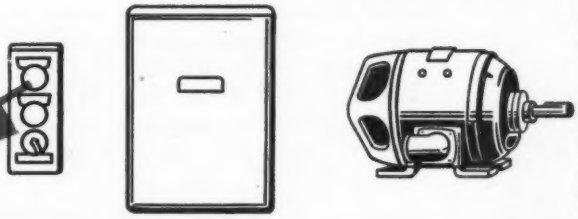
on small conveyors, blueprint machines, packaging machinery, lathes, drill presses, and many other types of power-driven machines. Type H1 Thy-mo-trol provides speed adjustment at the turn of a knob. Speed may be preset or changed during operation.

EASY TO INSTALL, SAVES SPACE

Only three components in this "packaged" drive—a control station, electronic control panel, and a specially designed d-c motor. No gears, belts or pulleys to adjust. Can be mounted on or near the machine. Easy to wire, operate, and service.

Fill in the coupon and send it to us for a copy of the bulletin describing this newest development in adjustable-speed drives.

Type H1 Thy-mo-trol for a small lathe. START-STOP push-button station mounted with the speed-adjustment knob. Cover removed from control panel to show rectifier tube and enclosed terminal case.



Apparatus Dept., Sec. B 676-282
General Electric Company
Schenectady 5, N. Y.

Please send me a copy of your new bulletin, GEA-5179, describing G-E Type H1 Thy-mo-trol.

NAME.....

COMPANY.....

ADDRESS.....

.....

TYPE H1 THY-MO-TROL . . . NEW LOW-COST DRIVE FOR EVERY SPEED

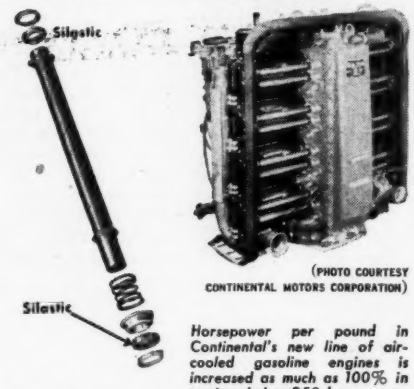
GENERAL ELECTRIC

Silicone News

DC SILICONES

More Power Per Pound†

Call it plain laziness or call it "divine discontent", some potent force has driven us to seek more concentrated motive power ever since man started to move on wheels. But each increase in horsepower per pound has increased operating temperatures and created a need for more heat resistant materials. Dow Corning Silicones help to supply that need in such major developments as the new air-cooled gasoline engine by Continental Motors Corporation of Detroit.



(PHOTO COURTESY CONTINENTAL MOTORS CORPORATION)

Horsepower per pound in Continental's new line of air-cooled gasoline engines is increased as much as 100% in engines below 250 hp . . . more than 300% (down from 2400 to 777 lbs.) in the horizontally opposed 8 cylinder 250 hp engine shown above. In this type of engine, Silastic gaskets are essential parts of the standard push-rod and tube assembly.*

Among the many problems solved by Continental engineers in designing this compact power plant, was the leakage of oil around the push-rod tube assembly. Hot oil and operating temperatures up to 500° F. severely limited the life of conventional gasketing materials. Continental engineers, however, knew about Silastic, the rubber-like Silicone by Dow Corning, and they asked for samples and technical assistance.

Silastic solved this problem because it is more heat-stable and more resistant to hot oils and to permanent deformation at temperatures above 200° F. than any other resilient material. Use of engines in sub-zero weather is not restricted by these gaskets of Silastic 161 because they are serviceable from —90° to 500° F. For more information about Silastic, ask for data sheet No. F-5B.

†See Silicone News, Dec. 1947, for story on 50% reduction in size and weight of electric motors made possible by Dow Corning Silicone Insulation.

*TRADE MARK REGISTERED U. S. PAT. OFF.

DOW CORNING CORPORATION
MIDLAND, MICHIGAN

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Los Angeles • New York
In Canada: Fiberglas Canada, Ltd., Toronto
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Dow Corning
FIRST IN SILICONES

Dow Corning Silicone Products include

FLUIDS
Damping
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Dielectric
Waterproofing
Lubricating
Diffusion Pump
Mold Release

GREASES
High Temperature
Low Temperature
Valve Lubricants
Stopcock
High Vacuum

COMPOUNDS
Ignition Sealing
Antifoam A

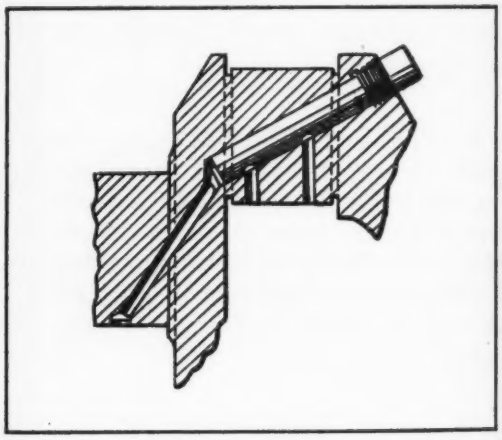
RESINS
Electrical Insulating
Laminating
Protective Coatings

SILASTIC*
Molding
Extruding
Coating
Laminating

*Trade Mark
Dow Corning
Corporation

set up excessive frictional resistance if externally supported by some sort of tubular metal sleeve. These problems are overcome by supporting the rubber sleeve with a series of ball-bearing mounted metal rings. These closely spaced rings turn freely relative to each other and allow a natural increasing angle of twist to take place in the rubber while supporting it against bursting pressures. The seal is designed to allow ready removal as a unit.

FOREIGN MATTER consisting of metallic particles or other solids are automatically trapped in the oil passage of a crankshaft before reaching the crankpin bearing surfaces with a new design covered in patent 2,440,812. Assigned to the Ford Motor Co. by Ernest L. Simpson, the crankshaft design utilizes a slanted, enlarged oil passage bore to feed oil under

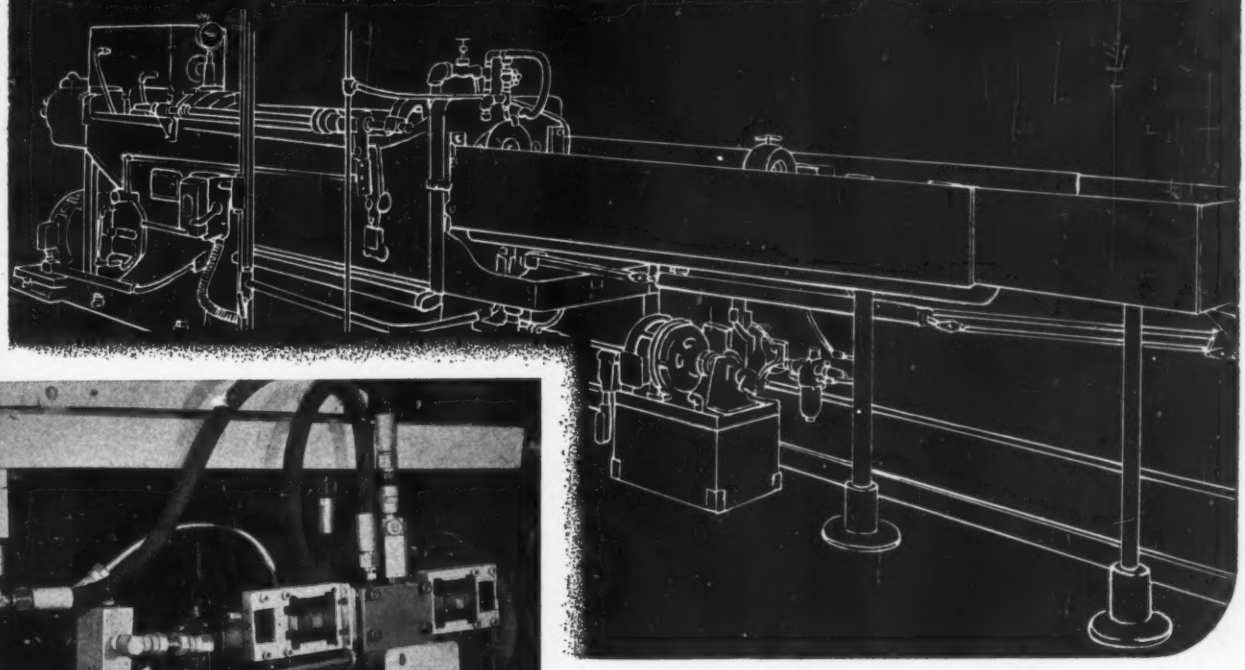


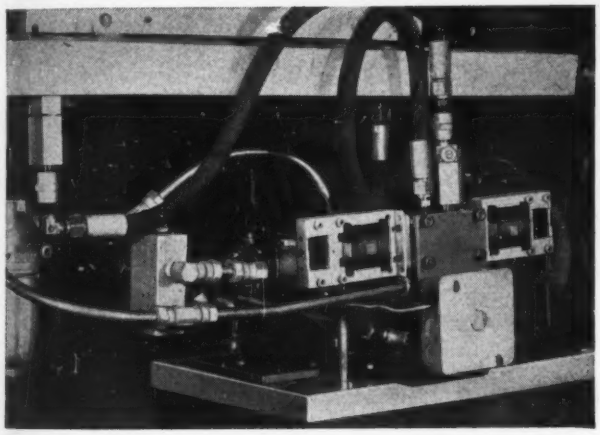
pressure to the crankpin bearings. Action of centrifugal force on particles of greater density than the oil causes them to be thrown against the outer wall of the bore where they accumulate and remain. The accumulated plug of sediment is removed by withdrawing a tubular cartridge provided for the purpose.

TWO-WAY ROLLER GUIDE covered in patent 2,444,432 provides within a single element, rollers to guide a slide both laterally and transversely. The unit, patented by William F. Eckert, is designed to have the main thrust roller situated within the lateral guide roller. Mounted on self oiling bushings, the unit requires a minimum of space for mounting.

POSITIVE LOCKING at both ends of its stroke is a feature of an unusual hydraulic actuator assigned to Electrol, Inc. by B. N. Ashton under patent 2,442,577. The piston may be released for actuation automatically by hydraulic pressure or manually by cam action in case of hydraulic breakdown or line failure. Manual unlocking action simultaneously releases relief valves in each head so as to allow free flow of fluid in either direction.

Broaching Machines, too, MADE BETTER... with Hydraulic Engineering by Electrol





The Electrol hydraulic installation which provides push button control for broaching tools.

GONE are the days when it was necessary for an operator to advance and return a broaching tool *manually* to obtain clearance space for the working piece.

Today, with Hydraulic Engineering by Electrol, this back-breaking task—often requiring the time and labor of two men—is eliminated simply and inexpensively by operating and controlling the broaching tool *hydraulically*.

This way . . . it requires only the pressure of a finger-tip on an electric push button to engage or disengage the broaching tool—or to move the tail stock in either direction from a stationary point.

Perhaps, the machines you use—or the products you make—can be made better through the use of hydraulic devices. Electrol's engineering staff will gladly work with you.

• • •

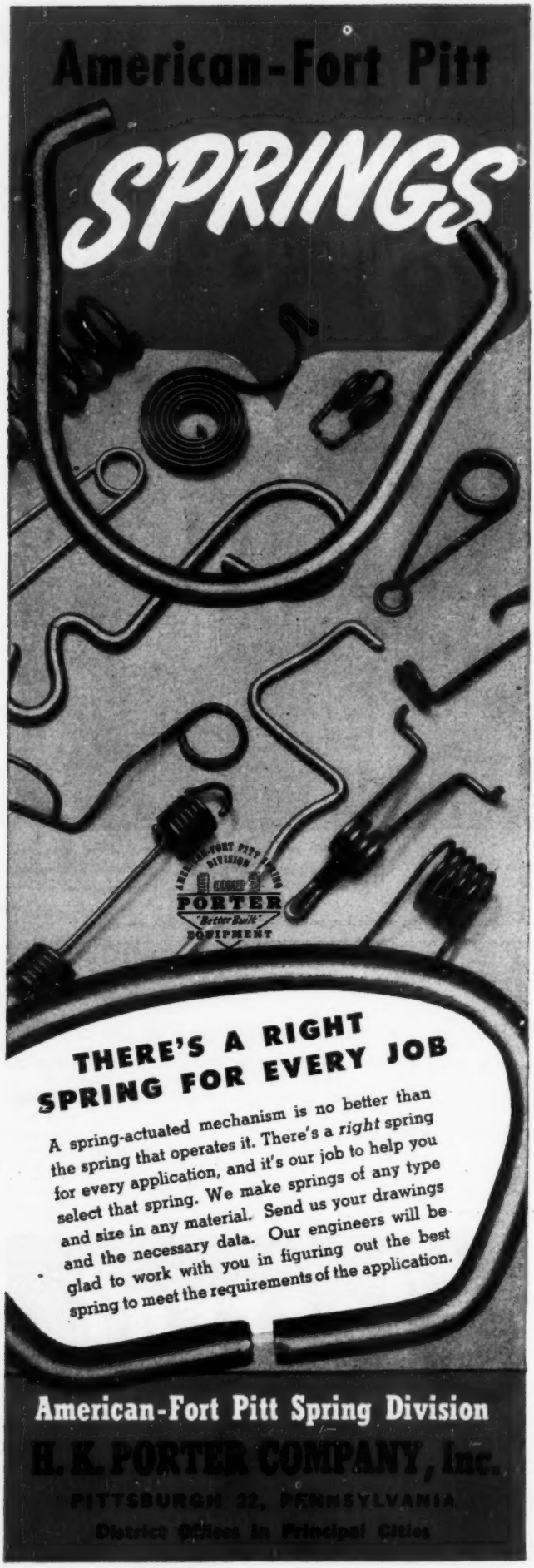
Electrol

INCORPORATED

KINGSTON, NEW YORK

CYLINDERS • SELECTOR VALVES • FOLLOW-UP VALVES
CHECK VALVES • RELIEF VALVES • HAND PUMPS
POWERPAKS • LANDING GEAR OLEOS • SOLENOID
VALVES • ON-OFF VALVES • SERVO CYLINDERS • TRANSFER
VALVES • CUT-OUT VALVES • SPEED CONTROL VALVES

FOR BETTER HYDRAULIC DEVICES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THERE'S A RIGHT SPRING FOR EVERY JOB
A spring-actuated mechanism is no better than the spring that operates it. There's a *right* spring for every application, and it's our job to help you select that spring. We make springs of any type and size in any material. Send us your drawings and the necessary data. Our engineers will be glad to work with you in figuring out the best spring to meet the requirements of the application.
American-Fort Pitt Spring Division
H. K. PORTER COMPANY, Inc.
PITTSBURGH 22, PENNSYLVANIA
District Offices in Principal Cities

Assets to a Bookcase

The Science and Engineering of Nuclear Power

Edited by Clark Goodman, Massachusetts Institute of Technology; published by Addison-Wesley Press Inc., Cambridge, Mass.; 500 pages, 7¾ by 10½ inches, clothbound; available through MACHINE DESIGN, *$7.50 postpaid.*

Based on a series of seminars held at the Massachusetts Institute of Technology, this volume is directed to those engineers interested in the industrial applications of nuclear energy. Consequently, it delves into the theory of nuclear physics and discusses the fission process and chain reaction. However, the preponderance of the text is devoted to actual engineering problems, for example the application of pile theory, control of piles, construction of nuclear reactors and the selection of nuclear fuels. A wealth of data is presented in chart and tabular form and numerous sketches illustrate the text.

□ □ □

Gas Turbine Construction

By R. Tom Sawyer, manager, research department American Locomotive Co.; published by Prentice-Hall Inc., New York; 369 pages, 6 by 9 inches, clothbound; available through MACHINE DESIGN, *$7.00 postpaid.*

Bringing together in one volume the latest facts on gas turbines, turbosuperchargers and jet engines, this text explains the fundamentals and discusses in detail problems of design and construction. It covers all types of units from the large stationary installations such as used with the Houdry process to the modern railway and aircraft turbines. American, British and German practice is described at length. Amply illustrated, the volume presents in chart, drawing, tabular, and photographic form a wealth of data which will be of value to the development as well as to the design engineer.

□ □ □

Precision Investment Castings

By Edwin Laird Cady, industrial management consultant; published by Reinhold Publishing Corp., New York; 356 pages, 6 by 9 inches clothbound; available through MACHINE DESIGN, *$6.00 postpaid.*

Of particular value to engineers concerned with the design of parts having complex shapes subject to extremely high stresses or made of refractory materials, this volume covers the field of investment

It's NEW - It's Sensational

The SMALLEST "O" or "I" Size Magnetic Made

ONLY 6 3/16" x 6 11/16" in size of enclosure — *half the size* of vertically-operated magnetics — this Type "RA" Starter is new throughout. Sturdy, dependable, *incomparably compact.*

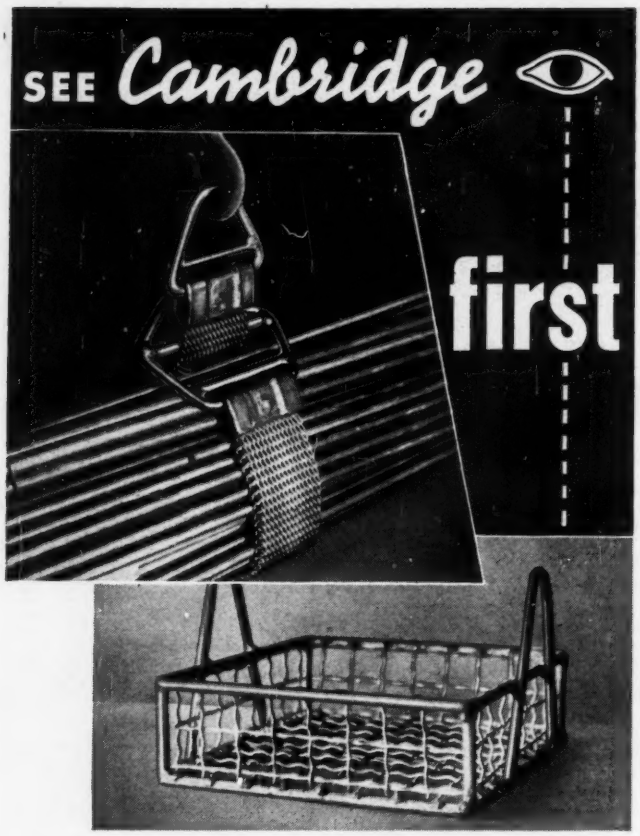
The *new principle* design with balanced mechanism provides a free-floating magnet with guided contact alignment. Lever and fulcrum action multiplies the leverage, transfers magnet pull from vertical to horizontal plane and increases contact pressure.

Designed for straight-through wiring, with separate contact enclosures. Embodies all the high-efficiency features found hitherto in the Size 3 and Size 2 "RA" Starters.

ARROW AH HART

This New *Size "0"* or *"1"*, Type "RA" provides local control for motors rated 5 H. P., 220 Volts and 7½ H. P., 440 or 550 Volts. Write at once for descriptive folder on this foremost advance in space-saving motor controls.

THE ARROW-HART & HEGEMAN ELECTRIC COMPANY, HARTFORD 6, CONN., U. S. A.



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Call in a Cambridge field engineer to discuss the proper type of fabricated wire units for your bulk handling operations. Let us show you how Cambridge Engineered wire fixtures are specially built from any metal and coating for any bulk operation—plating, dipping, spraying or heat treating.

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casting in such a manner as to be a worthwhile reference book for engineers. Of particular value to the designer will be the sections on economics of design, and accuracy of contours and the comparison of investment casting with other manufacturing processes. Of interest also will be the discussions of surfaces and design advantages to be realized in manufacture in large, medium and small quantities of parts. Balance of the book is largely devoted to production aspects covering mold design and materials, as well as casting techniques.

□ □ □

Powder Metallurgy

By Paul Schwarzkopf, president, American Electro Metal Corp.; published by the Macmillen Co., New York; 379 *pages,* 6 *by* 9¼ *inches, clothbound; available through* MACHINE DESIGN. $8.00 *postpaid.*

Possibly one of the most complete discussions of powder metals ever published, this volume should be of basic interest to the design engineer concerned with hard metals, complex shapes, friction parts, contacts, or magnets, etc. The author speaks with the authority of over thirty years' experience in the field, both in this country and in Europe; by virtue of his position as head of one of the largest powder metal companies in this country he has placed at the disposal of the reader much data not commonly available.

The book consists of three sections covering processing, products and theoretical principles. The first of these will give the engineer an excellent insight into the limitations and possibilities of the technique. Second section, dealing with the physical properties of powder metals, covers dense materials, porous products, refractory metals, hard metals, contact materials, and friction parts. In each of these the metallurgy and its effect on performance are discussed in detail. Third section of the book, dealing with theoretical principles, outlines the development of the theoretical concepts, and compares the properties of powder metals with metals produced by other processes.

□ □ □

Principles of Servomechanisms

By Gordon S. Brown, professor of electrical engineering, and Donald P. Campbell, assistant professor of electrical engineering, Massachusetts Institute of Technology; published by John Wiley & Sons Inc., New York; 400 *pages,* 6 *by* 9¼ *inches, clothbound; available through* MACHINE DESIGN, $5.00 *postpaid.*

Stressing the theory that the design of servomechanism systems requires direct mathematical analysis, this book outlines a creative synthesis approach to the problem. Belief that the common trial-and-error methods widely used are too costly and slow has re-

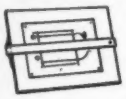
TWO ways you benefit from MB Isomode* Vibration-Isolators

1. IMPROVED VIBRATION CONTROL!

2. EASIER ENGINEERING!

Experiences of two well-known manufacturers demonstrate this double benefit:

Always on the alert to improve their product, a truck maker comprehensively tested Isomode mounts. Their adoption followed quickly. Because, instead of previous, typical truck characteristics, motors mounted on *Isomode* units displayed passenger-car performance! *Vibration was really isolated,* even though the units were not at optimum locations, but placed at standard points to allow interchangeability with earlier models.



Another company, with a tough vibration control problem because they use various makes and types of engines in their own product, discovered *engineering and production simplicity* through Isomode units. Vibration was controlled by units placed at the regular mounting points. This accomplishment is all the more remarkable when you consider that the vibration varies with each type of engine!

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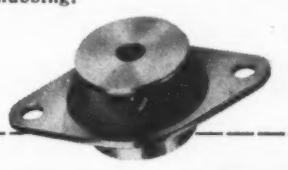

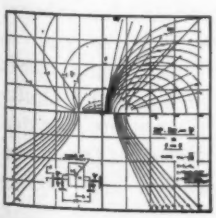

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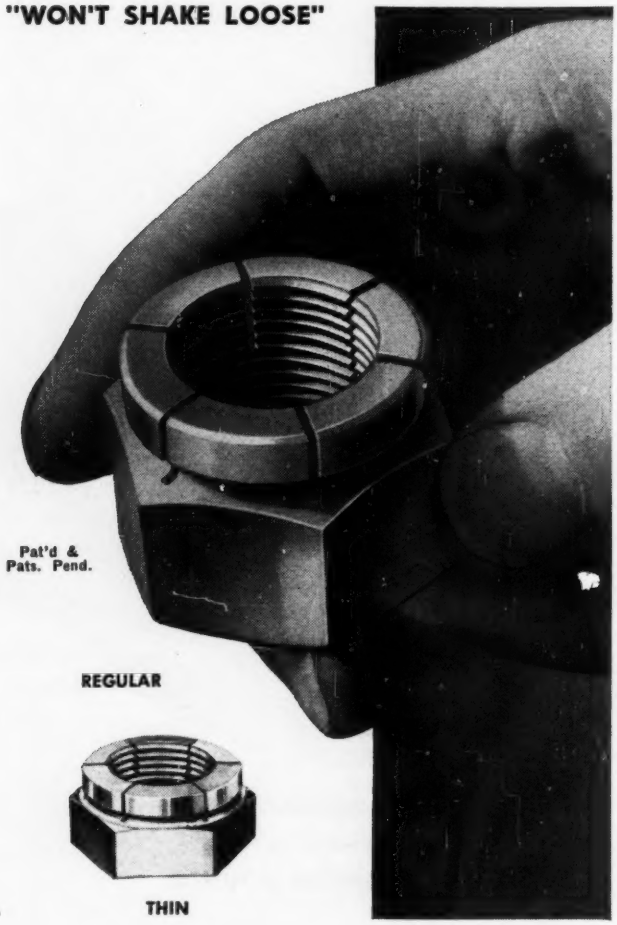

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Material presented is for the most part mathematical in nature. Equivalent circuits of both electrical and hydraulic systems are discussed. Both the transient and sinusoidal response methods are analyzed and tied in with the Laplace transform, the transfer function, and system behavior in terms of frequency spectra. While the mathematical approach has been stressed by the authors, sufficient material on practical experimental studies, together with numerous charts, diagrams, tables, and problems make the book a valuable reference for the design engineer.

□ □ □

Radio-Frequency Heating

By G. H. Brown and C. N. Hoyler, research engineers, RCA Laboratories, and R. A. Bierwirth, chief engineer, Sound Incorporated; published by D. Van Nostrand Co. Inc., New York; 370 pages, 6 by 9 inches, clothbound; available through MACHINE DESIGN, $6.50 *postpaid.*

Design and application of apparatus for induction heating are discussed in this timely volume. Written by experts in the field, the book covers its subject well, carefully developing the theory and then applying it in a number of typical designs. The first 15 chapters lay the groundwork for the subject, going into currents in conductors, coil efficiency and heat flow equations. The balance of the book is concerned with such applications as wood-glueing presses, radio-frequency sewing machines, radio-frequency heat treating, dehydration, etc. A basic book, it should be of particular value to those engineers just entering the field of high frequency heating.

□ □ □

Industrial Electronics Reference Book

Compiled by electronics engineers of the Westinghouse Electric Corporation; published by John Wiley & Sons Inc., New York; 680 pages, 9⅝ by 11⅞ inches, clothbound; available through MACHINE DESIGN, $7.50 *postpaid.*

This book deals for the most part with basic electronic application information rather than with specific apparatus that may soon be obsolete because of rapid technical advancement. Of particular interest to the design engineer, power rectifiers as well as radio frequency heating and electronic instruments are discussed along with their advantages and limitations in industrial use. The book also covers electronic circuit components including resistance forms, capacitors and oscillators. The operation and charac-

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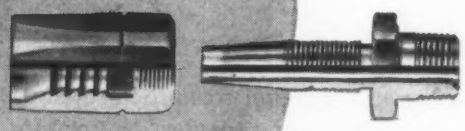

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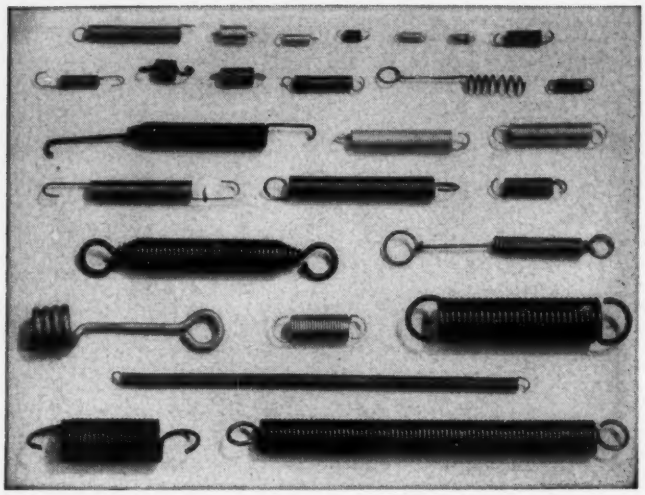

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teristics of different electronic tubes including vacuum, gas, X-ray, cathode, photoelectric, and ultraviolet tubes are explained. This well illustrated book is the combined work of 37 engineers, each thoroughly versed in his particular branch of the subject.

Complete listing of commercial and university testing and research laboratories has been compiled by the National Bureau of Standards under the name, "Directory of Commercial and College Laboratories." Information is given concerning 220 commercial laboratories with 80 branches and offices, and 189 college laboratories used for research and testing. The directory, identified as miscellaneous publication M 187, may be obtained for 30 cents from the Superintendent of Documents, Washington 25, D. C.

Recent National Bureau of Standards booklet by Brunetti and Curtis is said to be the first comprehensive treatment of printed electronic circuits ever published. The booklet consists of ten chapters totaling 43 pages and is illustrated with 39 illustrations and five tables. Six chapters cover methods of applying wiring and circuit components directly to an insulated surface in such a manner as to attain sturdiness and miniature size. These chapters are entitled: Painting, Spraying, Chemical Deposition, Vacuum Processes, Die-Stamping, and Dusting. Performance and application details as well as limitations are discussed in other chapters. The booklet is available at 25c from the Superintendent of Documents, U. S. Government Printing Office, Washington 25.

Report to the Navy department in booklet form entitled, "The Fracture of Metals," is a compilation of information on the laws and fundamental mechanism of fracture. This volume, written by M. Gensamer, E. Saibel, J. T. Ransom, and R. E. Lowrie, all of the American Welding Society, is divided into two parts. Part one includes a survey of the literature, an analysis of the theories of fracture and the application of principles. Part two discusses new developments in the theories of fracture and plastic flow. Bibliography provides over 300 references to the subject. This 84-page, paperbound booklet measures 6 by 9 inches and is available at $1.00 from the American Welding Society, 33 W. 39th St., New York 18.

Booklet entitled "Standard Welding Symbols" is the latest complete and authoritative list of symbols which may be used to indicate welds in engineering drawing. It has been carefully revised from previous manuals in order to achieve flexibility and also to cover all types of welding methods. The 69 page, 6 by 9-inch paper-bound booklet is available at 50 cents from the American Welding Society, 39 W. 39th St., New York 18.

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Protective Finishes for Springs

(*Continued from Page* 132)

than bad proportions. Many wire forms, flat springs, and other springs with hooks and various-shaped ends give extensive trouble in the plater's barrel. The designer should try to make changes which would reduce the tangling potential. In some cases, the closing of an eye to an opening smaller than the wire size will have the desired result, in others, the increasing of an opening will be the proper move. Wherever such changes can be made, or the contour of the part simplified, the designer may reduce the manufacturing cost of the part.

Tendency To Tangle Is Not Obvious

Examples of wire forms and springs subject to tangling are illustrated in *Figs.* 4, 5 and 6. Obviously, it is impossible to discuss all the design kinks which are capable of causing trouble, but the parts illustrated, and the comments pertaining to them, may suggest others. The tangling potential of a given part, especially when the part exists only as a list of specifications or a scaled outline on a drawing, is not at all easy to perceive clearly. Experienced platers and spring engineers often err in their judgment of this matter unless they have a dozen or so parts to handle.

There remain several other factors bearing on the plating of springs and wire forms from the proportion angle. Springs, *Fig.* 5, with straight ends or small hooks capable of slipping into or catching in the ⅛-inch holes in standard plating barrels will obviously cause difficulty. For large parts, suitable only for still-tank plating because of size, design should if possible take into account the fact that such parts must be racked. Therefore the cost of plating will be reduced where the design can include holes or hooks which will facilitate the quick and secure racking of the parts.

Plating Inside Surface Is Difficult

Extension springs, which are in most cases closely wound, present a difficulty that may not be apparent but should be understood by designers. Satisfactory plating cannot be obtained on inside surfaces (the metal being deposited travels in a straight line from the anode to the work, and whatever deposit occurs on the inside of a long close-wound spring is the result of more or less fortuitous chance). For this reason instructions are sometimes issued by government, as well as private buyers, specifying plating of the springs while under extension. The intention is to have the springs racked and extended so that there will be appreciable space between coils, permitting complete coverage of the wire by the plating metal. This result can be achieved at a high still-tank plating price, but the danger of damage to the spring is

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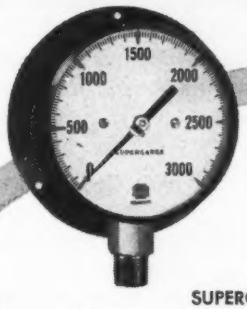

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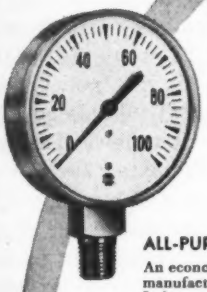

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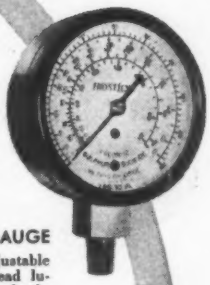

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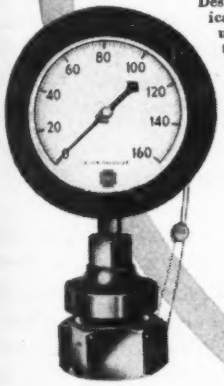

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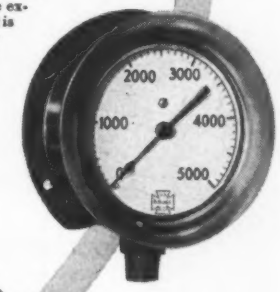

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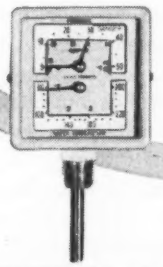

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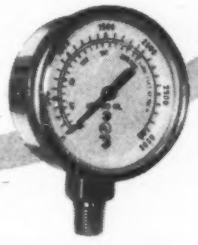

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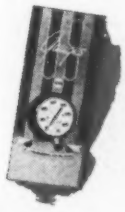

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so great that few platers will accept jobs having this requirement.

If complete protection for extension springs must be obtained, alternatives should be employed such as corrosion-resistant wire, preplated materials, or a phosphate-type coating, such as Parkerizing, which involves simple immersion rather than electrolytic action to achieve complete coverage.

Plating Causes Structural Deterioration

No electroplate improves a spring from the standpoint of function, except insofar as it protects the surface from corrosion and in this way lengthens its life. In other words if a spring is so applied that it is continually protected by oil, or is in a noncorrosive atmosphere, its life will be longer if not plated. The reasons for this are: (1) Some structural deterioration occurs in the material of all plated springs due to the hydrogen embrittlement caused by electrolytic action and, all too often, by the excessive use of acid in cleaning the springs before plating; (2) no matter how carefully a spring is cleaned, areas of imperfect adhesion occur. At these spots the molecular structure of the plating eventually breaks down and cracks. The way is then prepared for failure of the part through corrosion and fatigue of the material in the spring. It is therefore a good policy to specify oil or phosphate-type finish for springs wherever usage and customer acceptance will permit. This point is especially germane when fatigue loading is involved, as in automotive valve springs, and less important where stress is low, or the load static.

Use of Cleaner Is Important

Reference has been made to the cleaning of springs prior to the plating bath. Since this matter is of extreme importance, it should be stressed that, for carbon-steel springs in general, the cleaner used should contain 15 per cent caustic soda, 60 per cent ash, and the balance trisodium phosphate and wetting agent. Best results are obtained if the solution is pumped through the work with some force. Following this cleaner, a quick rinse (nothing more than an in and out dip) in a 5 per cent muriatic acid solution should be used to remove the cleaner from the parts. A minority of springs need to be put through a degreaser prior to the cleaning process. For springs made from copper-base materials the cleaner used should contain soda ash and trisodium phosphate, but no caustic soda.

Space does not permit a more detailed discussion of many of the points mentioned in this article, but it is hoped that this article may suggest a helpful approach to the problems facing the designer when he is required to decide on a finish for springs and wire forms.

Collaboration of the following organizations in the preparation of this article is acknowledged with much appreciation:

Accurate Plating Co.	Cleveland
Empire Plating Co.	Cleveland
Parker Rust Proof of Cleveland	Cleveland

MEETINGS AND EXPOSITIONS

Jan. 10-14—

Society of Automotive Engineers Inc. Annual meeting and engineering display to be held at Book-Cadillac Hotel, Detroit. John A. C. Warner, 29 West 39th St., New York 18, N. Y., is secretary and general manager.

Jan. 10-14—

American Society of Mechanical Engineers. Materials handling and management divisions meeting to be held at Convention Hall, Philadelphia. C. E. Davies, 29 West 39th St., New York 18, N. Y., is secretary.

Jan. 10-14—

National Materials Handling Show to be held at Convention Hall, Philadelphia, under the joint sponsorship of the Materials Handling Institute and the ASME management and materials handling divisions. Clapp and Poliak Inc., 350 Fifth Ave., New York 1, N. Y., is exposition management.

Jan. 24-27—

Institute of the Aeronautical Sciences. Seventeenth annual meeting to be held at Hotel Astor, New York. Robert R. Dexter, 2 East 64th St., New York 21, N. Y., is secretary.

Jan. 24-28—

American Society of Heating and Ventilating Engineers. Fifty-fifth annual meeting and ninth international exposition to be held at the International Amphitheatre, Chicago. A. V. Hutchinson, 51 Madison Ave., New York 10, N. Y., is secretary of the society and Charles F. Roth, Grand Central Palace, New York 17, N. Y., is exposition manager.

Feb. 13-17—

American Institute of Mining and Metallurgical Engineers. Annual meeting of the Iron and Steel and Institute of Metals divisions to be held in San Francisco. Additional information may be obtained from headquarters of the society, 29 West 39th St., New York 18, N. Y. Ernest Kirkendall is secretary of the metals divisions.

Mar. 8-10—

Society of Automotive Engineers Inc. Passenger car, body, and production meeting to be held at Book-Cadillac Hotel, Detroit. John A. C. Warner, 29 West 39th St., New York 18, N. Y., is secretary and general manager.

Mar. 28-30—

Society of Automotive Engineers Inc. Transportation meeting to be held at Statler Hotel, Cleveland. John A. C. Warner, 29 West 39th St., New York 18, N. Y., is secretary and general manager.

Apr. 11-13—

Society of Automotive Engineers Inc. Aeronautic and air transport meeting to be held at Hotel New Yorker, New York. John A. C. Warner, 29 West 39th St., New York 18, N. Y., is secretary and general manager.

Apr. 11-15—

American Society for Metals. Western metal congress and exposition to be held at Shrine Civic Auditorium, Los Angeles. W. H. Eisenman, 7301 Euclid Ave., Cleveland 3, Ohio, is secretary.

Apr. 25-28—

Southern Machinery and Metals Exposition to be held at the Municipal Auditorium, Atlanta. Michael F. Wiedl, 267 E. Paces Ferry Road N.E., Atlanta 5, Ga., is managing director.

Apr. 25-29—

American Society of Mechanical Engineers. Oil and gas power division conference to be held at Hotel Sherman, Chicago. C. E. Davies, 29 West 39th St., New York 17, N. Y., is secretary.

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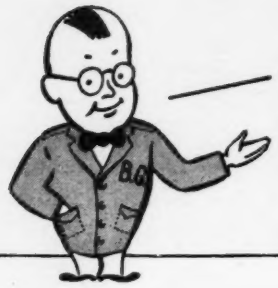

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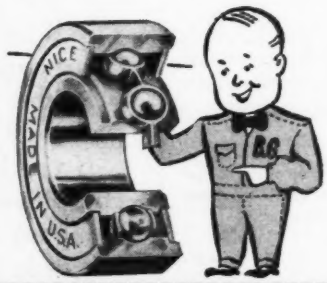
Wm. H. Taylor & Co., Inc.
ATLANTA 3, GA.
J. M. Tull Metal & Supply Co., Inc.
BALTIMORE 2, MD.
Carey Machinery & Supply Co.
BOSTON, MASS.
Bearings Specialty Company
Chandler & Farquhar Co., Inc.
BRIDGEPORT, CONN.
The Hawley Hardware Co.
BROOKLYN, N. Y.
H. L. Dickie
BUFFALO, N. Y.
Buffalo Bearings, Inc.
Root, Neal & Company
CANTON, OHIO
The Manufacturers Rubber & Supply Co.
CHARLOTTE, N. C.
Matthews-Morse Sales Co.
CHICAGO, ILL.
Bearing Jobbers, Inc.
Berry Bearing Company
Boston Gear Works
Chicago Pulley & Shafting Co.
Power Transmission Equipment Co.
Ray M. Ring Company, Inc.
CINCINNATI, OHIO
The Queen City Supply Co.
CLEVELAND, OHIO
Bearing Distributors, Inc.
Boston Gear Works
The Mau-Sherwood Supply Co.
The Ohio Ball Bearing Co.
J. A. Shomer Co.
DALLAS, TEXAS
Geo. J. Fix Company
DAVENPORT, IOWA
Standard Bearings Company
DAYTON, OHIO
The Klinger-Dills Company
DENVER, COLO.
Bearings Service & Supply Co.
The Western Belting & Packing Co.
DES MOINES, IOWA
Standard Bearings Company
DETROIT, MICH.
Detroit Ball Bearing Co. of Michigan
The Charles A. Strelinger Co.
EUGENE, ORE.
Woodbury & Co.
EVANSVILLE, IND.
Bearings Service Co., Inc.
FRESNO, CALIF.
Bearings Supply Co.
HARTFORD, CONN.
Silliter-Holden, Inc.
INDIANAPOLIS, IND.
Indiana Bearings, Inc.
Indianapolis Belting & Supply Co.
Vonnegut Hardware Co.
JACKSONVILLE, FLA.
The S. B. Hubbard Company
KANSAS CITY, MO.
Ellfeldt Machinery & Supply Co.
KNOXVILLE, TENN.
Hoyt N. Payne Co.
LOS ANGELES, CALIF.
Andrews Hardware & Metal Co.
Bearing Sales Company
Garrett Supply Company
Edward D. Maltby Co., Inc.
Pacific Bearings Company
LOUISVILLE, KY.
Industrial Equipment Co.
Kentucky Bearings Service, Inc.
MEMPHIS, TENN.
Lewis Supply Co.
MIAMI, FLA.
General Equipment & Supply, Inc.
MILWAUKEE, WIS.
The Western Iron Stores Co.
Wisconsin Bearing Company
MINNEAPOLIS, MINN.
R. C. Duncan Company
Minnesota Bearing Company
NEWARK, N. J.
Squier, Schilling & Skiff, Inc.
NEW ORLEANS, LA.
Industrial Ball & Roller Bearing Co.
McLellan A-C Supply Co.
NEW YORK, N. Y.
L. C. Biglow & Co., Inc.
Boston Gear Works
Frank Tracy, Inc.
OAKLAND, CALIF.
Bearing Specialty Company
C. W. Marwedel
PHILADELPHIA, PENNA.
Bearings, Inc.
Boston Gear Works
Lindsay, Oberholzer & Co.
Maddock & Company, Inc.
PITTSBURGH, PENNA.
Somers, Fitler & Todd Co.
PORTLAND, ORE.
Bearing Sales & Service, Inc.
Woodbury & Co.
PROVIDENCE, R. I.
Machine Parts Corporation
QUINCY, MASS.
Boston Gear Works
RICHMOND, IND.
The Queen City Supply Co.
RICHMOND, VA.
Apex Machine Mfg. Co., Inc.
ROCHESTER, N. Y.
John M. Forster Company
SACRAMENTO, CALIF.
The Bearing House
SALT LAKE CITY, UTAH
Bearings Service & Supply Co.
SAN FRANCISCO, CALIF.
Bearing Engineering Co.
C. W. Marwedel
SAN JOSE, CALIF.
Masterson Bearing Co.
SEATTLE, WASH.
Bearing Sales & Service, Inc.
Cragin and Company
SIOUX CITY, IOWA
Standard Bearings Company
SPOKANE, WASH.
Bearing & Rim Supply Co.
SPRINGFIELD, MASS.
Boston Gear Works
ST. LOUIS, MO.
Colcord-Wright Mchy. & Supply Co.
ST. PAUL, MINN.
R. C. Duncan Company
SYRACUSE, N. Y.
Syracuse Supply Co.
TAMPA, FLA.
Southern Pump & Supply, Inc.
TOLEDO, OHIO
The Ohio Belting & Transmission Co.
TRENTON, N. J.
Wiley-Hughes Supply Co.
TULSA, OKLA.
Allied Bearings Supply Co.
WATERBURY, CONN.
The White Supply Co.
WILKES-BARRE, PENNA.
Holmes Metallic Co.
WINSTON-SALEM, N. C.
Kester Machinery Co.
WORCESTER, MASS.
W. M. Steele Company, Inc.

"Now ready for Immediate Delivery. A new line of inexpensive "precision-type" ball bearings in Inch Sizes. Solid inner and outer races with ball retainer. Here's a rugged bearing that will give excellent service and save you money at the same time. Designed for medium loads and speeds up to 3000 RPM. Available from stock!"



FREE! Get this new ball bearing data book plus special folder on the new Boston-Nice "3000" series.

Name..... Position.....

Company.....

Street.....

City..... Zone..... State.....

One of the world's most complete lines
83 distributors in major cities

BOSTON GEAR WORKS
90 Hayward St., Quincy 71, Mass.

BUSINESS AND SALES BRIEFS

FORMERLY president of the Ward Leonard Electric Co., Dawson J. Burns has resigned that office and has been elected chairman of the executive committee. Taking the place of Mr. Burns as president of the company is Arthur A. Berard, previously executive vice president. Mr. Berard retains his post as general manager.

♦

Frank A. Colosimo has been appointed chief service engineer of the American Brakeblok Div. of the American Brake Shoe Co. He will be located at the headquarters of the division in Detroit.

♦

According to a recent announcement, Ralph M. Hoffman, president of the Link-Belt Co., Pacific Division, has been elected a vice president of the parent company. He continues as president of the West Coast organization.

♦

A Dayton, Ohio, office has been opened by the Lord Mfg. Co. The new office, located at 238 Lafayette St., is under the supervision of W. W. Dalton.

♦

Recently transferred to the Boston office of the Dow Chemical Co. from the New York branch, John F. White will be in charge of magnesium parts sales throughout the northeast part of the country. States covered will be Massachusetts, Vermont, New Hampshire, Maine and a part of Connecticut.

♦

With offices at 5225 Wilshire Blvd., Los Angeles, the A. W. Nash Co. will serve as California representative of Ajax Flexible Coupling Co. Inc. The Nash company succeeds Howard Speddy in handling the Ajax line, Mr. Speddy having retired from business.

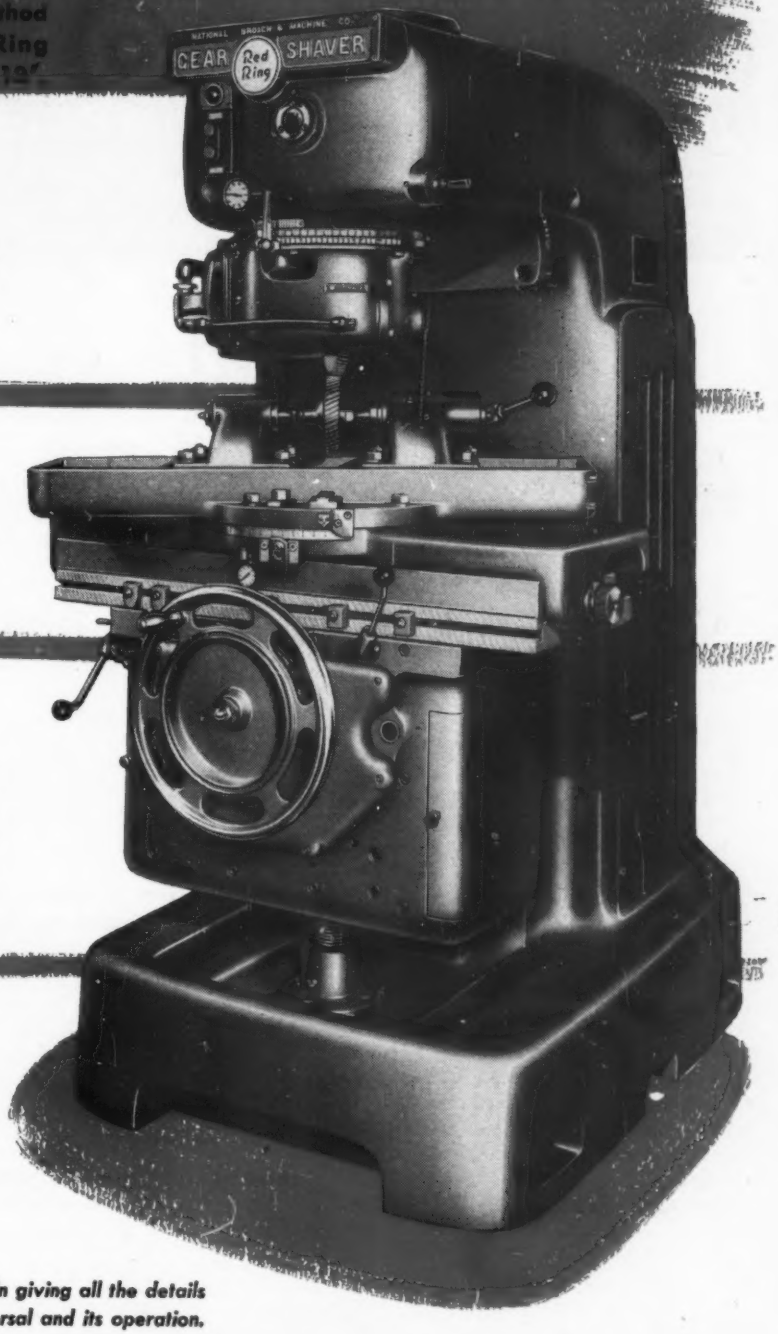
♦

New custom heat treating plant will shortly be opened in St. Louis by Lindberg Steel Treating Co. The new facilities, located at 650 E. Tay-

NEW GEAR SHAVING MACHINE

Has far greater production possibilities

Gears may be shaved by any method you choose on the new Red Ring Universal Model GCV-8" and 12"



conventional shaving

You can shave wide face gears by the conventional method of shaving, that is, when the gear is reciprocated across the cutter and parallel to its axis.

diagonal shaving

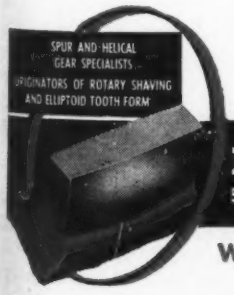
Or, you can shave the narrower face gears diagonally at much higher production rates and with considerably less cutter wear and less effort by the operator.

elliptoid tooth form

You can give the gear teeth the perfect Elliptoid form (crowning by rocking the table) while they are being shaved by either method mentioned above and without the necessity of using specially formed cutters. Special cutters may be used, however, if desired.

taper shaving

Likewise, gears may be taper-shaved by either the conventional or the diagonal method. This new machine is faster in operation and easier to load and unload than any of its predecessors.



Write for descriptive bulletin giving all the details of the new Red Ring Universal and its operation.

5003

NATIONAL BROACH AND MACHINE CO.
5600 ST. JEAN . . . . . DETROIT 13, MICHIGAN

WORLD'S LARGEST PRODUCER OF GEAR SHAVING EQUIPMENT

Corrosion and heat-resistant alloy steel castings of certain types acquire better characteristics when made by the Centri-die process in permanent molds. This method was developed by Firth-Vickers of Sheffield, England, over a long period of experimentation. It is largely responsible for the superior qualities of the Rolls-Royce, De Havilland, Bristol and other British airplane jet engines. Since the war it has been widely applied to castings for corrosion and heat-resistant service.

The agreement between Lebanon and Firth-Vickers makes available to us the best experience, methods and foundry practices known in England and assures Lebanon's customers a continuance of our traditional high-quality standards.

If your equipment is subjected to high temperatures or corrosive conditions, you should know about the new Lebanon castings made by the Centri-die process in permanent molds.

Get This Book
"Centri-die Centrifugal Castings"

Here is a clear explanation of the practical advantages to you of the Firth-Vickers Centri-die method of making alloy castings centrifugally in permanent molds. Of interest to executives and engineers who want to keep abreast of new manufacturing and production methods. Write for Bulletin G.

LEBANON STEEL FOUNDRY • LEBANON, PA.
"In The Lebanon Valley"

LEBANON *ALLOY AND STEEL* Castings CIRCLE L

lor Ave., will have 6000 square feet of floor space and will house 14 heat-treating furnaces.

♦

The Shops and Piston Ring divisions of the Koppers Co. Inc. have recently been combined to form a Metal Products division. All manufacturing activities of the new division will be in Baltimore under the direction of Walter F. Perkins, vice president and general manager.

♦

Los Angeles division of the Stamford Research Institute has been opened at 649 S. Olive St. It will function under the direction of Dr. A. M. Zarem.

♦

Manufacturing facilities of the Nice Ball Bearing Co., Philadelphia, have been increased by an enlargement of the company's Hunting Park Ave. plant.

♦

Assistant sales manager of American Wheelabrator & Equipment Corp. since 1946, Sherrill S. Deputy has recently been appointed sales manager.

♦

Appointment of P. J. Jensen, formerly executive engineer in the engineering and research department of Carboloy Co. Inc. to the position of manager of the Michigan district has been announced. Mr. Jensen, who will have offices in Detroit, succeeds M. E. Knoll who is retiring.

♦

According to a recent announcement, Nielsen Hydraulic Equipment Inc. has been appointed New York sales representative of the John S. Barnes Corp.

♦

New district office in Richmond, Va. has been opened by the Chain Belt Co. The office, located at 2900 W. Clay St., Richmond 21, will be under the direction of Fred W. Taylor.

♦

The American Standards Association recently became the American Standards Association Inc. through incorporation in the state of New York.

♦

Two-story building with approximately 34,000 square feet of floor space has recently been added to the Lincoln Engineering company's facilities in St. Louis.

♦

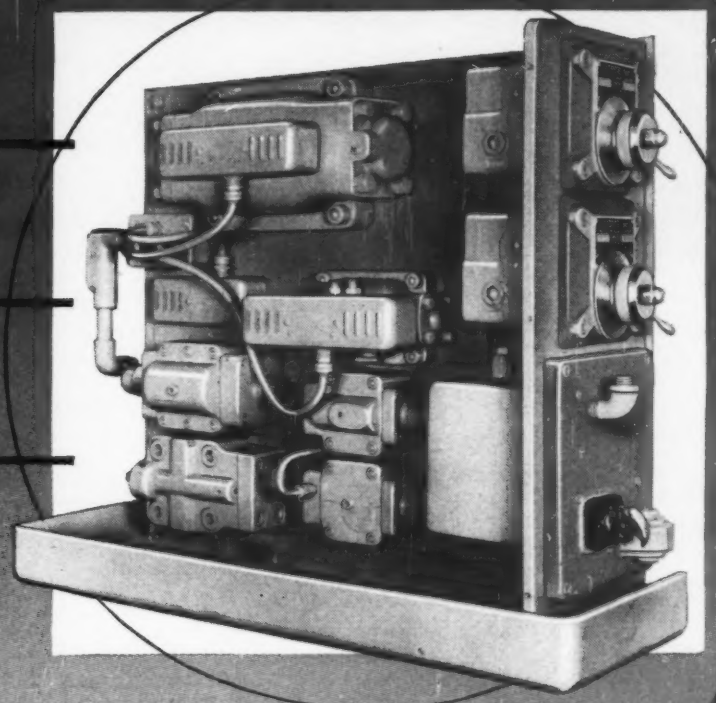
Formerly head of stainless-steel sales, western area, for Carnegie-Illinois Steel Corp., Frank B. Rackley

VICKERS Gasket Mounted Hydraulic CONTROL VALVES

Simplify Installation

Save Space

Make Adjustment Easier



This new Colonial Pull-Down Broaching Machine is an excellent example of how Vickers Gasket Mounted Valves simplify installation, save space and make adjustment easier for hydraulic control systems.

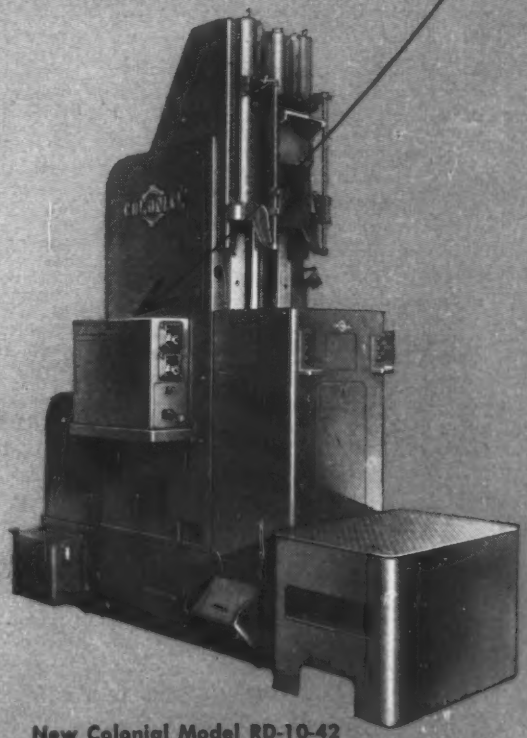
All valves in the panel box are easily accessible for adjustment by simply removing the cover. For cleaning and other maintenance, any valve can be completely removed by just loosening the hold-down screws . . . the piping is not disturbed and the system is not drained. The installation is simplified and more compact because all hydraulic piping connections are made into the opposite side of the machined surface upon which the valves are mounted with sealing ring type gaskets. The concealed piping also results in improved appearance.

Contact the Vickers Application Engineering Office nearest you for suggestions on how Vickers Hydraulic Equipment can improve your machinery.

All hydraulic control units for the machine are included in this group. Flow control valve adjustments convenient to operator without opening panel box.

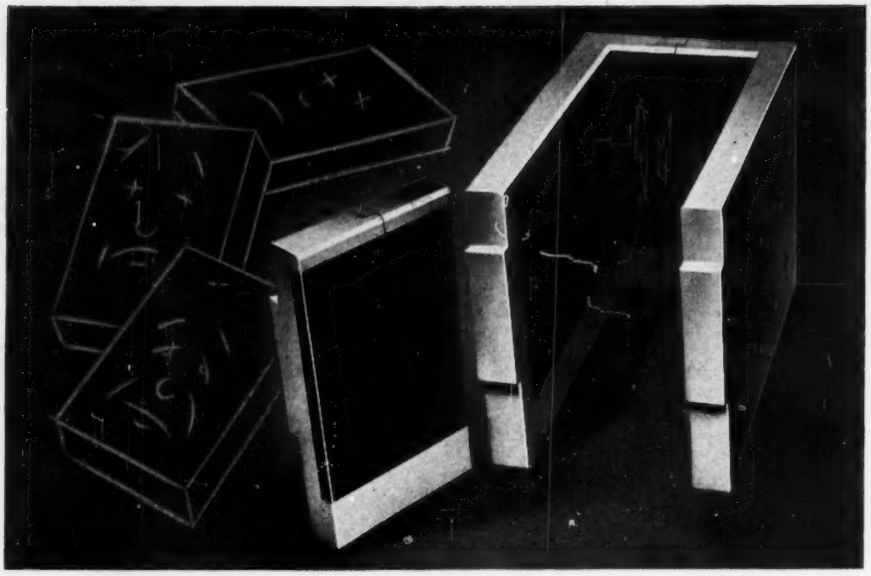
WRITE FOR NEW BULLETIN, 48-27

This bulletin will give you useful information regarding applications, advantages and installation drawings for Vickers Gasket Mounted Solenoid Controlled Directional Valves.



New Colonial Model RD-10-42 Pull-Down Broaching Machine has Vickers Hydraulic Controls.

VICKERS Incorporated
DIVISION OF THE SPERRY CORPORATION
1430 OAKMAN BLVD. • DETROIT 32, MICHIGAN
Application Engineering Offices: ATLANTA • CHICAGO • CINCINNATI • CLEVELAND • DETROIT
LOS ANGELES • NEWARK • PHILADELPHIA • PITTSBURGH • ROCHESTER • ROCKFORD
ST. LOUIS • SEATTLE • TULSA • WASHINGTON • WORCESTER



This brick mold won't lose face in a hurry!

MOLDS of hard steel and iron were rubbed the wrong way by coarse, abrasive brick. Excessive wear meant frequent replacement of brick-mold faces—and costs showed it!

Now? Now Carboloy Hard Metal faces are used on the molds—and this miracle metal is outwearing the old molds many times.

Surprising? Not to us, or to anyone who's worked with this *hardest metal made by man.* Carboloy's history of fantastic conquests over wear indicates that it could jump the life of some machine or product parts for you—by as much as a hundred times!

You see, *wear-proofing is our business.* Carboloy Cemented Carbides were *developed* to fight wear.

These qualities can go to work for you—to fight wear, and bring you lower costs and boosted production:

- **Extreme hardness**
- **High abrasion resistance**
- **Extreme density**
- **High modulus of elasticity**
- **Low coefficient of expansion**
- **High compressive strength**
- **Corrosion resistance**

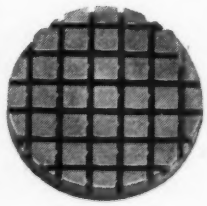
Check these points! Look over your methods and materials for places where wear might be too high. Better yet, call in a Carboloy engineer—let *him* seek out the spots where Carboloy Hard Metal can bring you real savings.

Choose from hundreds of low-cost standards

You'll find a standard to suit your need among Carboloy's huge, comprehensive stocks—it's almost a certainty. Carboloy's economical standards are fighting wear on many fronts. For example—

Here's an item as handy as a screwdriver to have around! Keep a supply of these unground solid Carboloy rods where you can use them in a hurry—for length gauges, flat drills, locating pins, all sorts of places and uses!

You'll never dig a fork into *this* waffle! A gauge anvil of Carboloy Cemented Carbide, used on measuring instruments, brings real savings by reducing the number of rejects due to faulty inspection equipment.



CARBOLOY COMPANY, INC.
11113 E. 8 Mile Street, Detroit 32, Michigan

CARBOLOY®
CEMENTED CARBIDE

HARD METAL
TO REDUCE WEAR ON MACHINES AND PRODUCT PARTS

has been appointed general manager of sales for Jessop Steel Co. He succeeds T. W. Pennington, former vice president, who resigned recently.

♦

The Rapids-Standard Co. Inc. of Grand Rapids, Michigan, has adopted the term "RapiStan" as a trademark to designate all equipment manufactured in their line.

♦

With offices at 1920 Union St., Oakland 7, Calif., the American Brass and Copper Sales Co. has been named distributor for special aluminum mill products manufactured by the Reynolds Metals Co.

♦

Frederick J. Ellis has recently been appointed assistant merchandising manager, pumps and compressor sales for Worthington Pump and Machinery Corp.

♦

Announcement has been made by the J. P. Marsh Corp. of the acquisition of the Witt Gauge Div. of National Pressure Cooker Co. The purchase will broaden the line of gages manufactured by the parent company.

♦

Several appointments have recently been made in the New Departure Division of General Motors Corp. The first of these is that of Frank J. Miller to the post of general sales manager of the division. Also, Lorne F. Lavery, formerly manager of the Detroit office, has been made assistant general sales manager at Bristol, Conn., while Charles D. McCall, formerly manager of automotive sales at Detroit, succeeds Mr. Lavery as manager of the Central Region. Concurrently, Howard A. Offers, sales engineer, has been appointed manager of the Mid-western Region with headquarters in Chicago, and Raymond J. Lynch is now supervisor of engineering for the central and Mid-western Region.

♦

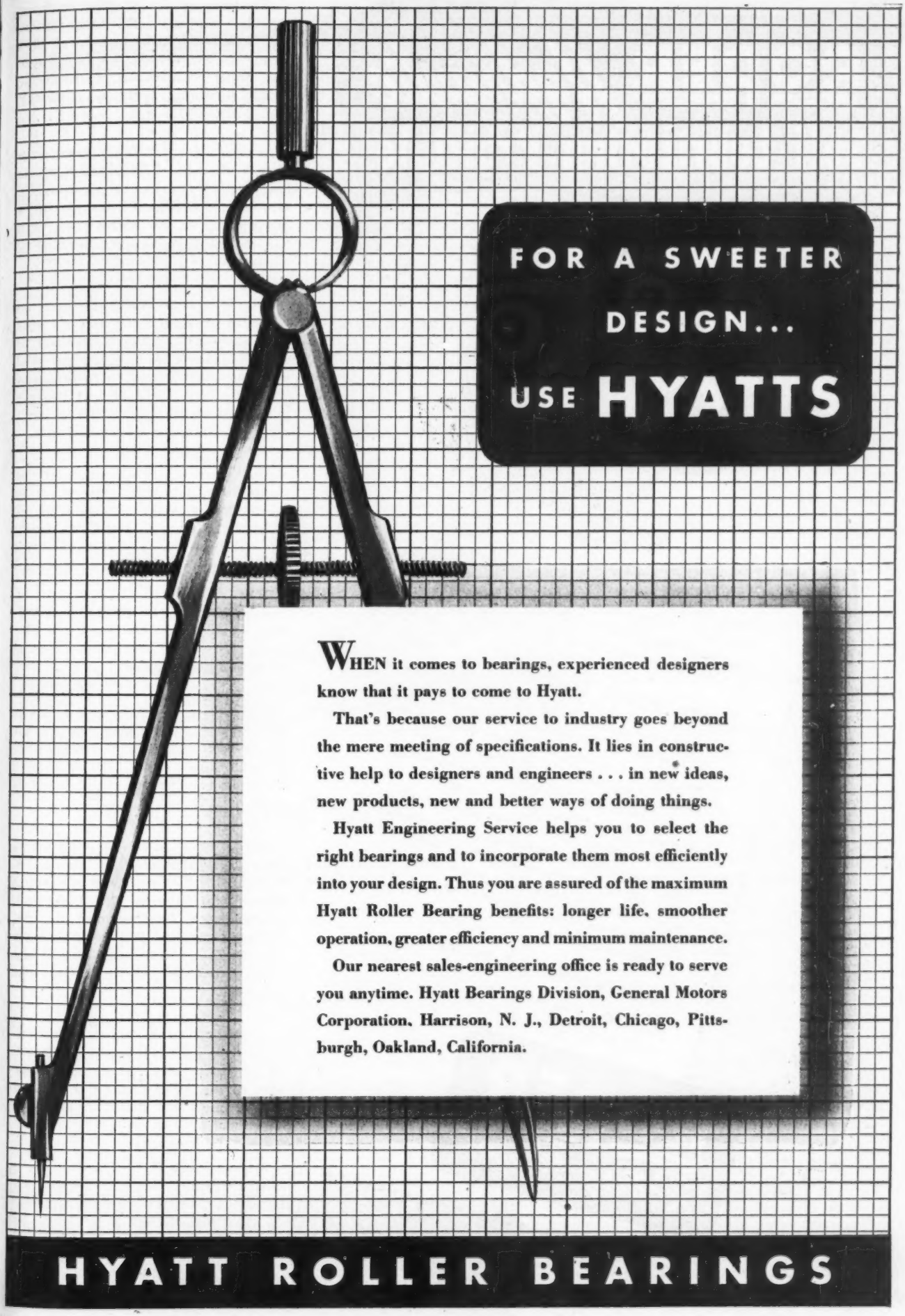
High precision aircraft parts will be developed and manufactured in a new $2,650,000 combined plant and laboratory to be built by the Pesco Products Div. of the Borg-Warner Corp. The new facilities will be located in the Cleveland area.

♦

Atlanta office of the United States Rubber Co. has recently been expanded by the construction of a new building at Lee and Donelly Sts.

♦

The Cambridge Wire Cloth Co. has announced the appointment of Paul



WHEN it comes to bearings, experienced designers know that it pays to come to Hyatt.

That's because our service to industry goes beyond the mere meeting of specifications. It lies in constructive help to designers and engineers . . . in new ideas, new products, new and better ways of doing things.

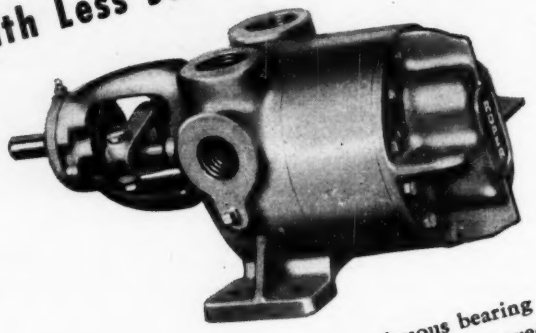
Hyatt Engineering Service helps you to select the right bearings and to incorporate them most efficiently into your design. Thus you are assured of the maximum Hyatt Roller Bearing benefits: longer life, smoother operation, greater efficiency and minimum maintenance.

Our nearest sales-engineering office is ready to serve you anytime. Hyatt Bearings Division, General Motors Corporation, Harrison, N. J., Detroit, Chicago, Pittsburgh, Oakland, California.

HYATT ROLLER BEARINGS

ROPER Hydraulic Self-Lubricating Principle

Prolongs Pump Life with Less Service Attention



Roper Pumps have positive, continuous bearing lubrication. The liquid pumped enters grooved bearings from the pressure side and is drawn to the suction side through grooves in the opposite bearings.

OTHER ROPER FEATURES

Floating equal size external gears, supported at both ends with high-grade bronze bearings, and choice of packed box or mechanical seal units to suit individual job requirements, efficient operation in either direction, easy accessibility to working parts cuts down "out-of-service" time. Four-port design offering eight optional piping connections cuts installation time and cost. Four piping arrangements are possible with pump operating clockwise . . . four other arrangements with pump operating counter-clockwise.

WIDE RANGE OF SIZES

Capacities ¾ to 300 g.p.m. — pressures up to 1000 p.s.i. — speeds up to 1800 r.p.m.

Send for Bulletin of Facts Today

SEE OUR CATALOG IN SWEET'S

ROPER Rotary Pumps

GEO. D. ROPER CORP., 252 Blackhawk Park Ave., Rockford, Ill.

V. Thomure as district sales engineer for the company's St. Louis office. Mr. Thomure will be located at 1248 Syndicate Trust Bldg., and will cover the states of Missouri, southern Illinois, southern Indiana, Kentucky and Tennessee.

♦

Appointment of John Frazier as New York sales representative of the Sterling Mfg. Co. has been announced. Mr. Frazier will handle the company's instruments in the entire state with the exception of metropolitan New York.

♦

Among the recent promotions in the staff of the Ex-Cell-O Corp. was that of John F. Miller to the post of sales manager of the machine tool and the cutting tool divisions. At the same time, D. H. McIver was appointed sales manager of the aircraft and miscellaneous parts divisions.

♦

The Houston, Texas, office of the Foxboro Co. has been moved to 2518 South Boulevard, Houston 6. The office will be housed in its own building which will also provide facilities for servicing of Foxboro equipment.

♦

Appointment of Charles H. Kaufman as district manager of its Chicago office has been made by the United States Gauge Co.

♦

Buffalo branch office of the Crocker-Wheeler Electric Mfg. Co. has recently been reopened. Located in the Ellicott Square Bldg., the office will be under the supervision of Louis E. Rau.

♦

Three new sales representatives have been appointed by the L.G.S. Spring Clutch Corp. They are: Smith Power transmission Co., Cleveland; Transmission Engineering Co. Inc., Philadelphia; and Potter and Dugan Inc., Buffalo. Potter and Dugan also have suboffices in Rochester and Syracuse.

♦

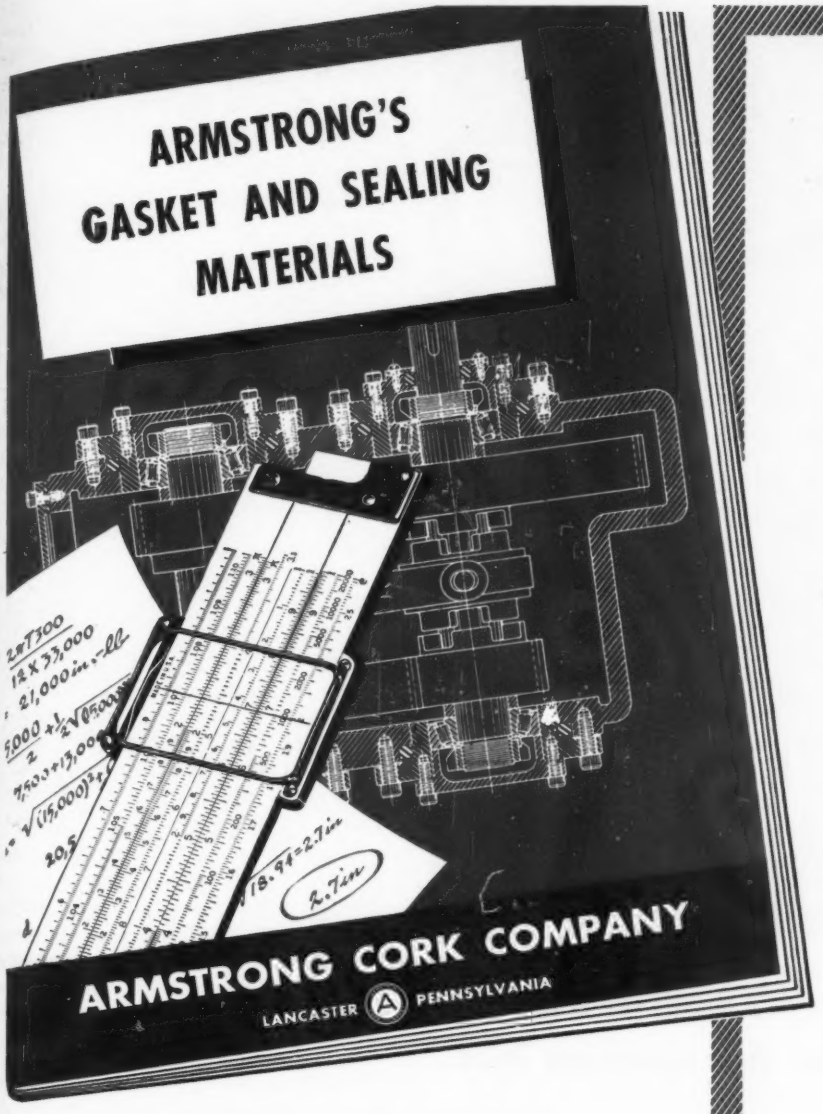
According to a recent announcement, O. W. Schmidt has been appointed Central States district sales manager of the Atlas Chain and Mfg. Co. Mr. Schmidt will maintain offices in Chicago.

♦

Representing C. P. Clare & Co. in Northern California and Nevada, will be E. L. Berman Co., 1355 Market St., San Francisco 3.

♦

The Variety Machine and Stamping Co. of Cleveland has been pur-



Send for helpful GASKET GUIDE

If you design, specify, or buy gaskets, you'll find this 24-page gasket guide helpful. "Armstrong's Gasket and Sealing Materials" shows typical applications. It suggests methods of putting Armstrong's stock materials to specialized uses in such fields as the radio, electrical, automotive, petroleum, and transportation industries. It also includes government specifications—plus up-to-date charts on synthetic rubber, cork-and-synthetic-rubber, cork composition, and fiber sheet sealing materials. For your free copy, fill in coupon and mail today.

Table of Contents

- Cork Compositions
- Cork Specialties
- Cork-and-Synthetic-Rubber Compositions
- Fiber Sheet Packings
- Synthetic Rubber Compounds
- Stock Materials That Do Specialized Jobs
- Designing Flanges for Efficient Sealing
- Designing Gaskets to Reduce Cost
- Effect of Gasket Width on Compression
- Effect of Surface Condition on Gaskets
- Factors in Choosing Resilient Gaskets
- Proper Compression for Resilient Gaskets
- Relation of Gasket Thickness to Load
- Resistance to Chemicals
- Sealing Oils with Synthetic Rubber
- Sealing with Confined Resilient Gaskets
- Tolerances for Gaskets

ARMSTRONG'S
GASKETS • PACKINGS • SEALS

Armstrong Cork Co., Gaskets and Packings Dept.
5112 Arch Street, Lancaster, Pennsylvania

Please send me at once a copy of the new 24-page booklet, "Armstrong's Gasket and Sealing Materials."

NAME_____

COMPANY_____

ADDRESS_____

CITY_____ STATE_____

Centrifugally-cast Ampco Metal valve-seat inserts are quality features of the 61 and 74 cu. in. O.H.V. engines on the famous Harley-Davidson motorcycles.

Durable Ampco Metal gets the hot seat . . .

—because it has the wear-resistance to "take it" on the toughest jobs

When a famous motorcycle manufacturer specifies Ampco Metal for valve-seat inserts *he knows his customers are getting a "plus" value.* And that value is featured by the manufacturer in advertising literature as an important sales appeal!

That's logical when you consider the long life and trouble-free service you gain, when critical parts are made of Ampco metal—with its outstanding resistance to corrosion, compression, impact, fatigue, and wear. It has excellent bearing qualities, too, plus unique efficiency at extreme temperatures.

Call your nearby Ampco engineer for full information on Ampco Metal and Ampcoloys . . . available in castings, extrusions, sheet, forgings, and fabricated assemblies. Write for latest literature.

Ampco Metal, Inc.

Department MD-12 • Milwaukee 4, Wisconsin

Field offices in principal cities

chased recently by Wallace F. Ardussi, sales and engineering executive. Mr. Ardussi was formerly vice president of Hupp Corp., in charge of sales and engineering development.

♦

Formerly manager of the Chicago district of the Allis-Chalmers Mfg. Co., B. F. Bilsland has been promoted to manager of the company's newly formed Midwest region.

♦

Appointment of Lewis P. Naylor as sales manager of the Ozalid Div. of General Aniline & Film Corp. has been announced. Mr. Naylor was formerly eastern sales manager of the Victor Adding Machine Co.

♦

Two new sales offices have been opened by Presstite Engineering Co. An east coast office with James W. Bampton in charge has been opened in the Times Medical Bldg., Philadelphia, while on the west coast Harry A. Lowry has opened a sales office at 5960 Willowcrest Ave., North Hollywood.

♦

Special division has been created by the E. B. Wiggons Oil Tool Co. Inc. to assist manufacturers in the development of quick disconnect couplings for use with their products. James A. Roberts heads up this new special-equipment division.

♦

The Atlas Sponge Rubber Co., manufacturer of molded and sponge rubber products, has recently been acquired by Roy N. Phelan. Location of the organization is 806 E. 61st St., Los Angeles 1.

♦

A number of appointments have recently been made by Allegheny Ludlum Steel Corp. H. N. Arbuthnot, formerly assistant to the president of the corporation, has been transferred to Washington, D. C. where he will serve as representative to the various governmental agencies. In the sales department, Truman B. Brown has been appointed assistant district manager of sales for the Detroit territory while Rodger S. Ahlbrandt, former district manager of sales for the Pittsburgh territory, has been named to succeed Mr. Brown. Max J. Pischke has been appointed to take the place of Mr. Ahlbrandt in the Pittsburgh office.

♦

Hydraulics division of the Airex Mfg. Co. has recently been acquired by Manning Maxwell & Moore. The

DESIGNED TO ECLIPSE *the performance of all previous carbide razor blade dies!*

"NEVER BE SATISFIED", is a sound production and profit policy which the Barth organization has practiced for a quarter century.

The carbide razor blade die pictured above is a typical example.

With predecessor dies already producing almost unbelievably sensational results, Barth designed this new and latest carbide razor blade die to eclipse all previous performance records . . . millions of hits with practically no sign of burr will result with this die which will be fed at the rate of 400 units per minute in the mechanical feeding device illustrated, which also recoils the strip stock after passing through the press.

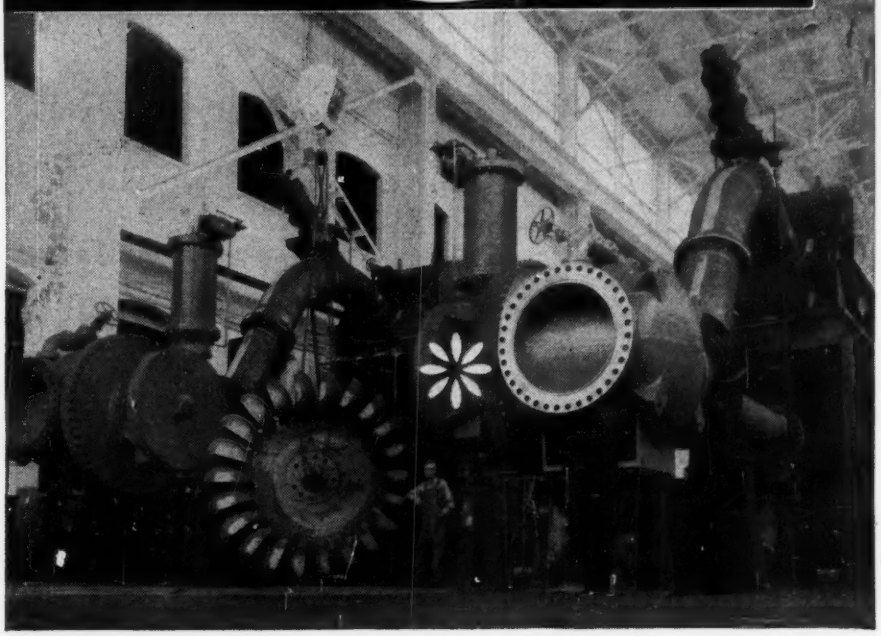
Results like these pay off in lower production costs, which in turn mean greater profit with no increase in price.

If you have a tool, die or production operation with which you are not completely satisfied, Barth has the experience, the engineers and the equipment to get better results for you. Ask for the Barth catalogue.

BARTH STAMPING & MACHINE WORKS Inc.
Denison at W. 34th St., Cleveland 9, Ohio

DESIGNERS AND BUILDERS OF DIES, TOOLS AND SPECIAL MACHINERY

CASTINGS BY *Continental*



KNOW THIS *Sure* SOURCE WHEN YOU DESIGN FOR CASTINGS!

Know that you can build what you design! Know Continental as a *sure* source when you design for castings! Continental's foundries and machine shop facilities are keyed to large, heavy and intricate steel castings of custom design as well as production runs from standard patterns. You can be sure of obtaining dependability and economy in any type of casting . . . carbon or alloy steel . . . heat treated, quenched and drawn . . . rough or full finish machined . . . in sizes up to 250,000 lbs. Continental facilities are your *sure* source for castings. For problem-solving counsel on castings, submit your early plans to Continental. No obligation, of course!

Designed by S. Morgan Smith Co., York, Pa. . . . all steel castings by Continental . . , for a giant hydroelectric plant built by Cia. Chilena de Electricidad, Ltda., Chile, S.A. A typical co-operation between designer and Continental!

Design with an eye to supply!

Continental

Continental
FOUNDRY & MACHINE CO.
CHICAGO . PITTSBURGH
Plants at E. Chicago, Ind.; Wheeling, W. Va.; Pittsburgh, Pa.

new owner will develop industrial applications for Airex hydraulic products in addition to the present line of Airex hydraulic valves and filters.

♦

Robert N. Griffith has been appointed manager of the Indianapolis district sales office of Berger Mfg. Div. of Republic Steel Corp. His territory covers Indiana and part of Kentucky.

♦

New officers were recently elected by the Society of the Plastics Industry. They are as follows: Chairman of the board, Neil O. Broderson; president, George H. Clark; vice president, Gordon Brown; secretary, Norman Anderson; treasurer, J. J. B. Fulenwider.

♦

Organization of the Superweld Corp. with Walter T. Wells as chairman of the board of directors has recently been announced. Capitalized at $500,000, the organization will engage in electric copper-brazing of steel, localized-heat silver brazing, bright annealing and the manufacture of specialized metal products.

♦

According to a recent announcement Romec Pump Co. has been acquired by Lear Inc. on a stock-exchange basis. Present executives will remain at Romec, becoming members of the Lear organization.

♦

Appointment of M. A. Sievert as an engineer in the technical service department of the Reynolds Metals Co. has recently been announced.

♦

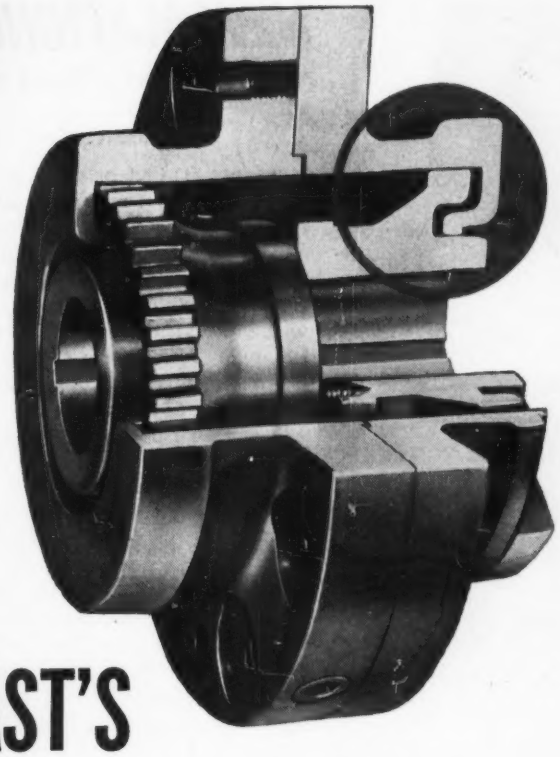
Ralph A. Mitchell, vice president of the Pittsburgh Forgings Co., has been elected president of the Drop Forging Association.

♦

With offices in Detroit, Harvey G. Moore is now assistant district manager of the Michigan area for the Wayne Pump Co. He was formerly sales manager with the Lord Mfg. Co.

♦

Merger of the Food Machinery Corp. and Westvaco Chemical Corp. was recently announced. Food Machinery Corp. will change its name to Food Machinery and Chemical Corp. and the business of Westvaco will be operated under the direction of its present management as the Westvaco Chemical Div.

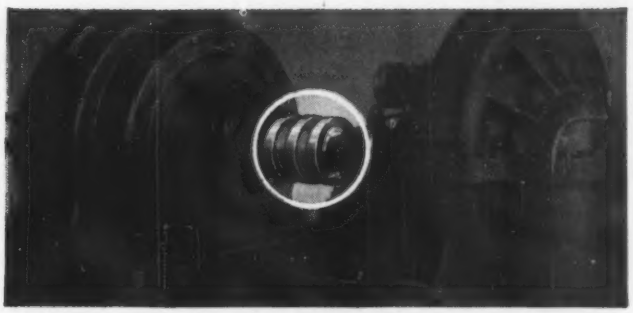


FAST'S

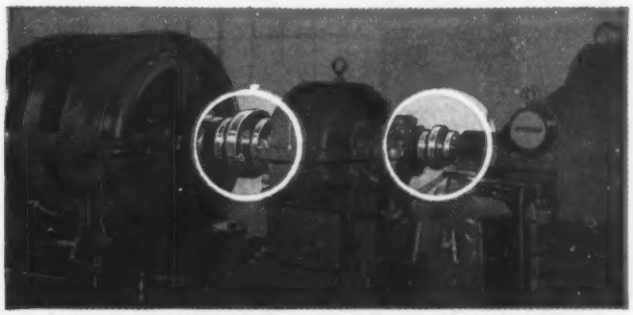
ALL-STEEL Construction Gives NON-STOP Power Transmission!

These installations have an important feature in common. They're all protected against coupling shutdowns by Fast's self-aligning Couplings. That's because these original gear-type couplings are all-steel throughout, *without perishable parts!* Even the load-carrying oil is protected by an *exclusive* "rocking bearing." This bearing is *exclusive* in providing a positive metal-to-metal seal which keeps out moisture, dust and grit. The bearing is also *exclusive* in its correctly engineered position which allows *freedom of movement* to compensate for misalignment because its *spherical* base has *the same axis* as the hub spline faces. No perishable packing rings are used in this vital spot.

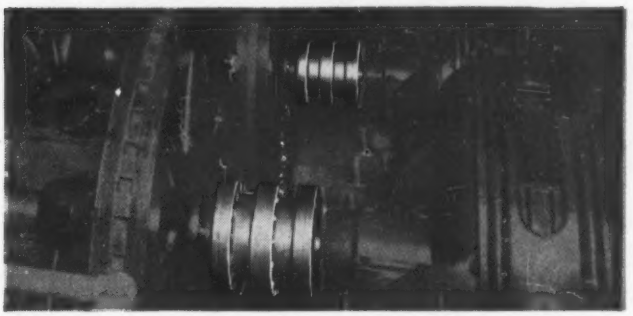
We have a complete line of couplings for immediate delivery. No matter which type of Fast's Couplings you buy, you get the benefits of years of top engineering experience, Koppers' high standards of workmanship and unexcelled coupling service which assure you of spare parts no matter how old your couplings may be. That means non-stop power transmission, longer machine life, lower costs, minimum shutdown losses! Koppers Co., Inc., Fast's Coupling Dept., 352 Scott St., Baltimore 3, Maryland.



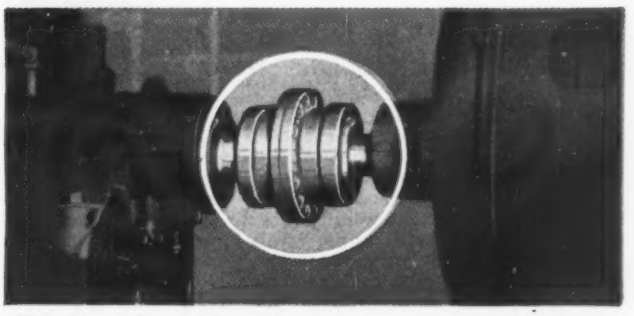
Fast's Coupling connects 1,800 h.p., 600 r.p.m. motor to 30 million-gallon-per-hour pump at City of Montreal pumping station.



These No. 3 and No. 5 Fast's Couplings connect a turbine-reduction gear-generator hookup, 1125 h.p. at 3740/1200 r.p.m., at Firestone Steel Products Co., Akron, Ohio.



This group of Fast's Couplings has been giving trouble-free service since 1922 at the Western Maryland Dairy Company's Baltimore plant.



This No. 4 Fast's Coupling connects a motor and induced draft fan at the West Penn Power Company's Beach Bottom, W. Va., plant.

K KOPPERS

FAST'S *self-aligning* COUPLINGS

SEND FOR FREE CATALOG giving detailed descriptions, engineering drawings, dimension and capacity tables and typical installation photographs for the many types of Fast's Couplings. Fill out this coupon and mail it to: Koppers Co., Inc., Fast's Coupling Dept. 352 Scott St., Baltimore 3, Md.

NAME.....POSITION.....

COMPANY.....

STREET.....CITY.....STATE.....

5 Kennamatic tools in action, roughing a forged steel stem pinion, SAE 5140. Speed is 230 SFM, feed .027", depth of cut 3/16" . . . and the Kennamatic tools turn out 350 pieces per index, 56,000 pieces per tool!

Another typical example of time-saving Kennamatic tooling. This cast iron valve body is turned at 250 SFM, with a feed of .005", depth of cut from 1/8" to 3/16". On this job other brazed tools had to be changed 8 times per shift . . . whereas Kennamatic inserts are simply indexed twice per shift, and give four days' service before re-sharpening.

When the going gets tough . . . . Kennametal Tooling is the sure way to faster production at less cost

Why wait for the going to get tough?

Under the pressure of rising costs—it becomes "tougher" and "tougher" to produce at a profit, and sell at a price.

But remember that "tough" is a relative term—any job is tough if there's an easier, faster, less expensive way to do it.

Your machining operations are tough, slow, and expensive . . . unless you are using uniformly strong, hard, long-lived Kennametal coupled with the latest Kennametal improvements in carbide tool design.

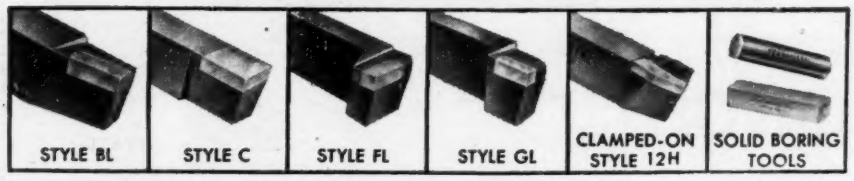
Kennametal tooling keeps your machines on the job with minimum interruption—enabling you to turn out from 2 to5 times more than steel tools on the same machine in the same time, with from 5 to 20 times longer tool life.

Use this modern tooling technique to save money in your plant—ask our district tool engineer to demonstrate.

Write for new Catalog 48.

KENNAMETAL *Inc.* **Latrobe, Pa.**

MANUFACTURERS OF SUPERIOR CEMENTED CARBIDES AND CUTTING TOOLS THAT INCREASE PRODUCTION



NEW MACHINES

And the Companies Behind Them

Commercial

DUPLICATOR. Direct-process, hand-operated. Prints four colors at once by changing carbons when making master copy. Requires no ink, stencil or gelatin. 300 copies from master typed or drawn on glossy paper with special carbon; 60 per min. Range, postcard size to 8½ by 14 in. Includes supply of carbon master units, fluid, and copy paper. American Photocopy Equipment Co., Chicago.

INDUSTRIAL VACUUM CLEANER. Portable. Four sizes: 2, 3, 5, and 7½-hp motors. Explosion proof if desired. Static water lifts, 60 to 105 in. Air velocities, 315 to 480 mph, moving 200 to 360 cfm. Self-cleaning air filter. Air flow reversible for blowing out motors, etc. Two intakes equipped with 1½ or 2-in. hose. Breuer Electric Mfg. Co., Chicago.

AUTOMATIC ELECTRIC DISHWASHER. Simplified drain system reduces installation cost. Self-sealing aluminum door requires no gaskets or springs. One-piece cabinet and 30 per cent fewer parts. Available in under-the-counter, free standing, or 48-in. sink-attached units. All have work surface area, top spray, front opening, and electric drying. Hotpoint Inc., Chicago.

Communications

SOUND PICTURE PROJECTOR. Portable, light-weight unit adequate for audiences to 100. Capable of projecting 52 by 70-in. picture. Reel capacity, 2000 ft; size, 8 by 11½ by 15½ in.; weight, 26 lb. Movie-Mite Corp., Kansas City, Mo.

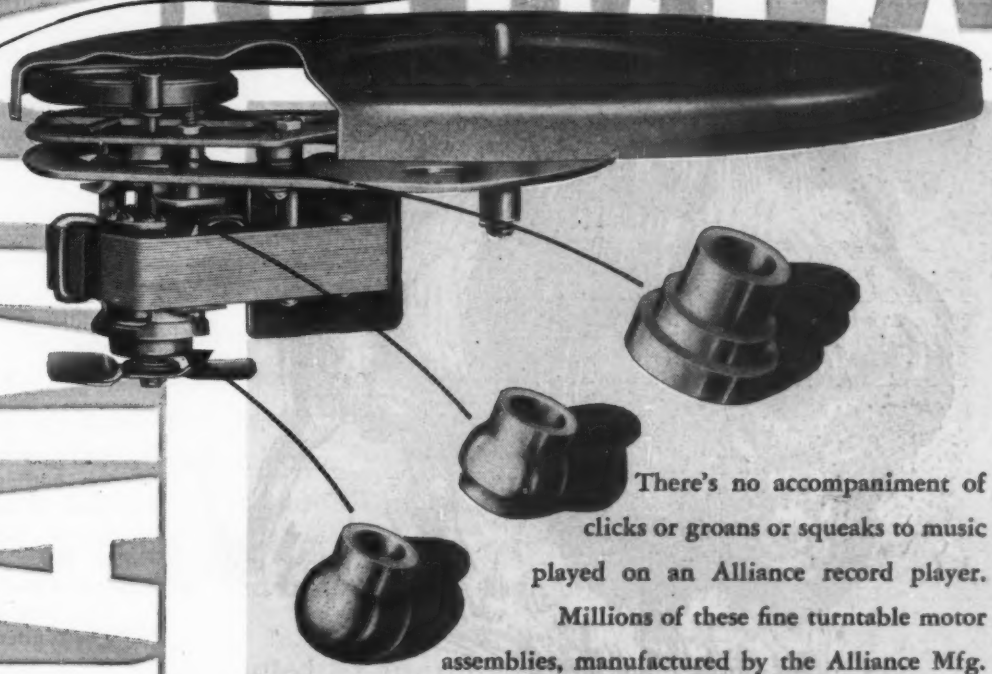
SOUND MOTION PICTURE PROJECTOR. Portable, single-case projector. Six-inch speaker mounted on removable door in side of case. Door may be swung out for use, or removed from case and placed near speaker. Ten-watt amplifier permits use of larger speaker if desired. Silent and sound speeds, reverse and still-picture clutch included. Bell & Howell Co., Chicago.

Domestic

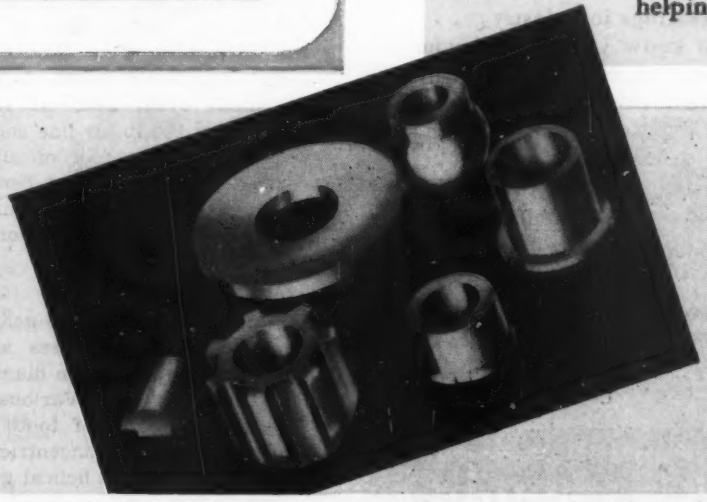
ELECTRIC RANGE. Pushbutton control. Heating element goes directly to indicated speed without going through several other switch positions. Cam interlock on switch contacts prevents closing of more than one circuit on any switch at the same time. Three models; double ovens, single ovens, built-in pres-

GRAMIX

GRAMIX bearings help insure quiet operation of record player



There's no accompaniment of clicks or groans or squeaks to music played on an Alliance record player. Millions of these fine turntable motor assemblies, manufactured by the Alliance Mfg. Co. of Alliance, Ohio, operates *smoothly, silently —and at constant speed*—thanks largely to three little Gramix bearings. Two self-aligning bearings in the motor and a double flange bearing on the main shaft are not only helping Alliance units give better service but they are also helping Alliance cut their cost of manufacture.

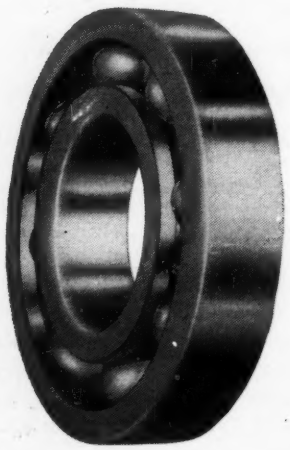


Because bearings, gears and other Gramix parts are die-pressed of powdered metal, they cost far less than machined parts, *sometimes as much as 80% less.* And because they can be impregnated with lubricants, further lubrication of Gramix parts is often unnecessary. Our engineers may be able to show how Gramix can help you improve mechanical performance and cut manufacturing costs in your products. Write today for free copy of the 264 page Gramix catalog.

THE UNITED STATES GRAPHITE COMPANY • SAGINAW, MICHIGAN

SWIFT • SURE • STURDY • SILENT • SMOOTH • STEADY

PLAN *Sturdy* BISON-TOUGH CONSTRUCTION IN EVERY BASIC DESIGN



You've often remarked that no piece of equipment can be superior to its component parts...

You appreciate the broad recognition and acceptance of BCA Ball Bearings in industry . . .

You know you can tell your engineers to draw upon our half-century of experience in friction control practices, encompassing many varied fields of industry . . .

You, who insist upon continuous, full-capacity, trouble-free operation, know this can be assured by specifying BCA Ball Bearings.

BEARINGS COMPANY OF AMERICA
LANCASTER, PENNSYLVANIA

BCA RADIAL • ANGULAR CONTACT • THRUST BALL BEARINGS

sure cooker and various surface unit sizes. General Electric Co., Bridgeport, Conn.

Earth Moving

COMPACTION ROLLER. Ground pressure varied from 218 to 670 psi by filling drum with water and loading attached boxes with sand. 144 tamping feet on drum, each with 8 sq in. tamping surface. Overall length, 16 ft 9½ in.; width, 13 ft 7½ in.; height, 6 ft 4½ in. Weight (empty), 14,000 lb; (loaded), 42,883 lb. R. G. LeTourneau Inc., Peoria, Ill.

EARTH MOVING TRACTOR. For high-speed, small-yardage projects. Self-loading, one-man operated. Powered by 100-hp diesel or gas engine. Top speed, 25 mph. Used with 9-ton or 7-cu yd scraper. All scraper operations controlled from operator's position. Torque proportioning differential automatically supplies power to wheel on firmest footing. Air brakes; 14 by 32 tapered bead tires; 19-ft turning radius. R. G. LeTourneau Inc., Peoria, Ill.

Industrial

DUST COLLECTOR. Portable. For collecting dust and lint from grinding, buffing or polishing operations. Size, 12 by 14 by 18 in.; weight, 43 lb. Rated at 320 cfm at 1½ in. water lift. Self-clearing paddle wheel fan driven by 1/3-hp motor. Has dust collecting tray and spun-glass filter. Several inlet connections available. Aget-Detroit Co., Ann Arbor, Mich.

PORTABLE AIR COMPRESSOR. Self-contained, air-cooled, gas-engine powered. Two-cylinder, single-stage compressor of 37.5 cfm piston displacement. Automatic unloader permits unit to run at constant speed but compress air only when needed. Pneumatic tires, trailer hitch attachment for towing. Ingersoll-Rand Co., Phillipsburg, N. J.

AIR IMPACT HAMMER. Variable power and speed air press. Impact adjustable from 2800 to 40,000 lb. Requires 100-lb air line connection; uses one cubic foot of air per 15 strokes. Hand or foot control. Capacity, 60 strokes per min. Blank Machine Tool Co., Jackson, Mich.

Inspection

GEAR CHECKER. Checks gears, hobs, worms, milling cutters and form tools to 10-in. outside diameter and 8-in. face width. Various fixtures permit checking of tooth spacing, pitch radius, concentricity and taper of spur and helical gears, and parallelism and crowning of spur gears. Special indicators check other

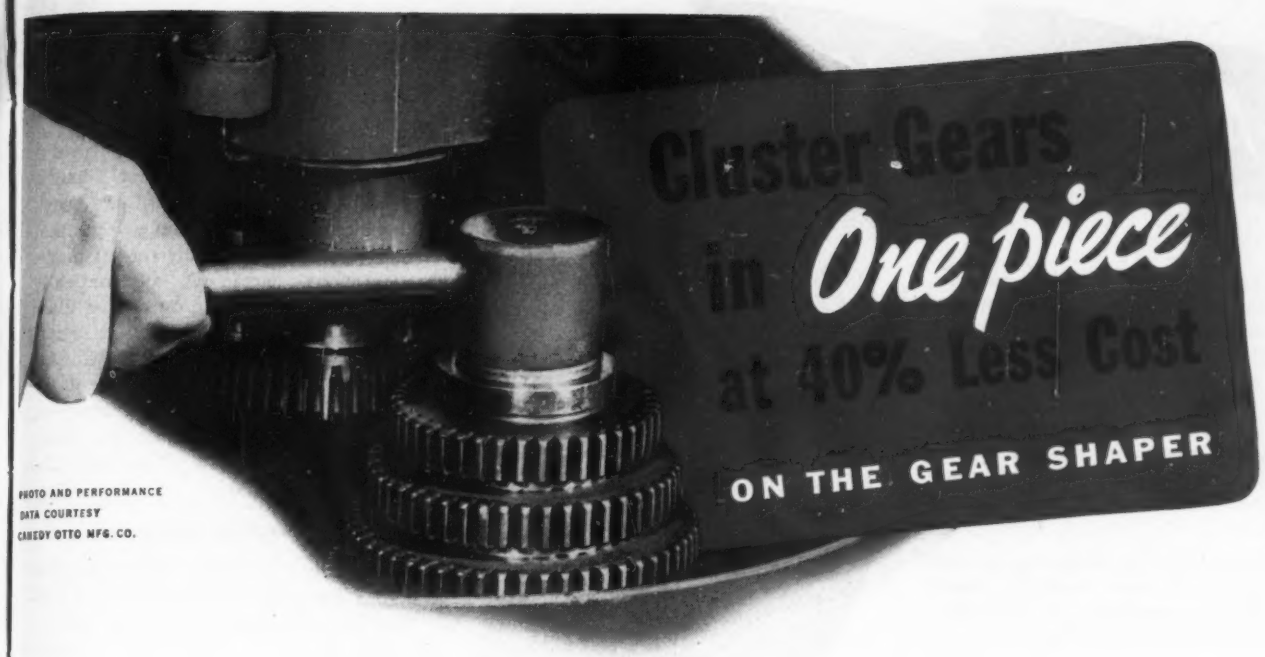


PHOTO AND PERFORMANCE DATA COURTESY CARDEY OTTO MFG. CO.

MACHINE DESIGNERS have long recognized the advantages of *one-piece design* for cluster gears, and other integral unit parts. The Gear Shaper has made the production of such designs possible to a greater degree of accuracy, and on a more economical basis.

One-piece cluster gears, generated on the Gear Shaper, eliminate misalignment in the fitting and keying of assembled gears. The number of machining operations is decreased, and overall costs reduced.

The versatility of the Gear Shaper opens the door to the practical production of many intricate shapes... increases the scope of low-cost machining at the designer's command. Our illustrated book, "The Art of Generating with a Reciprocating Tool" will prove of interest to every production-minded design engineer.

Reciprocating Generation
THE BASIS OF UNPARALLELED GEAR SHAPER VERSATILITY
12 Variable Motions and Positions
Plus Unlimited Cutter Designs
A CUTTER THAT RECIPROCATES WITH (OR WITHOUT) ROTATION
WORK POSITIONED IN ANY PLANE ...WITH OR WITHOUT ROTARY, LATERAL OR COMBINED MOTIONS
A CUTTER SPINDLE THAT CAN BE MOVED IN AND OUT
CUTTER PROFILES OF ANY REQUIRED CONTOUR

Fellows

THE FELLOWS GEAR SHAPER COMPANY

Head Office and Export Dept., Springfield, Vermont. Branch Offices: 616 Fisher Bldg., Detroit 2, 640 West Town Office Bldg., Chicago 12, 7706 Empire State Bldg., New York 1.

THE FELLOWS METHOD...MACHINES AND TOOLS FOR ALL OPERATIONS FROM BLANK TO FINISHED GEAR

NONE FINER!

High precision, automotive type gears produced by Fairfield are given the benefit of every modern facility and skill for producing FINE GEARS. This is why many makers of machine tools . . . agricultural implements . . . powered vehicles . . . construction machinery . . . printing presses . . . engines . . . motors . . . stokers and pumps regularly depend on Fairfield for their gears.

If you use gears in the products you build, check with Fairfield. Submit your gear problems for FAIRFIELD ENGINEERING SERVICE.

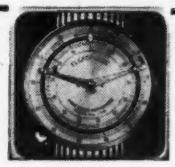
Fairfield Manufacturing Co.
311 So. Earl Avenue, Lafayette, Indiana

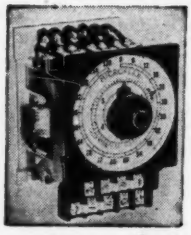
FAIRFIELD for FINE GEARS

EAGLE TIMERS for Controlling Industrial Processes

REPEAT CYCLE

Use where ON-OFF operation is continuously repeated. The ON time and OFF time are each adjustable on the dial. (FLEXOPULSE)





MICROMETER DIAL

Provides exceptional timing accuracy where a circuit is to close or open with a time delay. Timing adjustable over wide range.

(MICROFLEX TIMER)

COUNTER

Use for limiting a process to an exact number of operations. The counter contact opens after 1 to 400 electrical impulses as selected on dial. Automatic spring reset.

(MICROFLEX COUNTER)

MULTIPLE CIRCUIT —ADJUSTABLE

Use where several circuits are to close in a predetermined sequence. Time of closing and opening each circuit is adjustable.

(MULTIFLEX TIMER)

To reduce costs and improve quality of your products by automatic TIME-COUNT control . . .

- Write for catalogue Bul. 291.
- Send for details of your control problems to Eagle for recommendations.
- Consult Eagle representative in principal cities.

EAGLE Signal Corp.
MOLINE ILLINOIS

inspection work. Cast iron base provides dustproof storage for indicators. Adjustable headstock and tailstock. Michigan Tool Co., Detroit.

BLADE CHECKER. Inspects turbine blades and compressor vanes. Blades checked for contour, twist, thickness, and width in one minute; permanent inspection record made. Rotating scriber automatically traces blade outline on coated glass plate. Master inspection chart compared with scribed plate. Attachments measure amount of deviation of twist, displacement, thickness, or contour profile. Universal Engraving & Colorplate Co., Inc., Buffalo.

Manufacturing

DEGREASER. Two-dip, hand operated unit for cleaning medium-sized metal parts. Heated by steam, gas or electricity, or combined gas-and-steam or gas-and-electric. Immersion-solvent-vapor process. Capacity, 1000 lb steel parts per hr. Max. part size, 18 by 21 by 12 in. Floor space, 10 sq ft; weight, 500 lb. Detrex Corp., Detroit.

PLASTICS INJECTION PRESS. Manual, semiautomatic or automatic operation. 2-oz capacity; can be equipped to handle Nylon or vinyl materials as well as all other thermoplastics. Three-way heat nozzle heater furnished on Nylon unit. Mold assembly clamps at any angle between horizontal and vertical. Hydraulic mold closing and injection. Mold size, 8 by 9 in.; thickness, 6 in.; mold opening, 6 in. Standard injection pressure, 16,250 psi; can be adapted to 26,000 psi. Moslo Machinery Co., Cleveland.

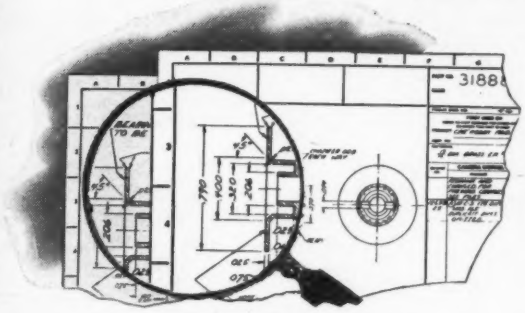
CAN MAKING MACHINE. Automatic body maker for dry package work or sift-proof talcum type cans. "Mennen type" seam. Square, rectangular, round, or oval can bodies. Automatic feed; lithographed or plain sheets. Electrically controlled. 300 fpm on round cans using plain seams. Weight, 5850 lb; floor space, 56¼ by 87¼ in. Lima Hamilton Corp., New York.

METAL CLEANING MACHINE. For wet abrasive cleaning and finishing of metal surfaces. Removes scale, rust and metal particles by projecting fine abrasive suspended in water against surface. Single blast-nozzle model, 30 by 30 by 77 in.; 1 to 8-nozzle model, 48 by 42 by 108 in. Blast nozzles only wearing parts. Armstrong Chemical and Machine Co., Painesville, Ohio.

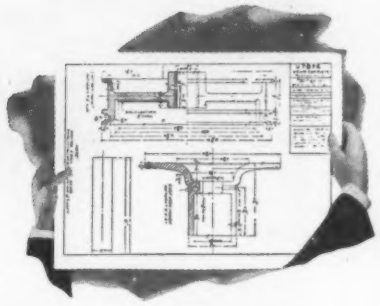
ARC WELDERS. Both ac and dc lines. Features of ac line: built-in power-factor correction, stepless current

To get intermediates as good as originals—and better

. . . specify this new photographic paper



Gets more out of every drawing . . . Intermediates or second originals? That's work "made to order" for Kodagraph Autopositive Paper. Silver-sensitized, photographic, it "steps up" contrast—prints to brilliant ink-line density from originals of every type—from new or old, weak-line drawings . . . from opaques, blueprints, direct process prints. And it does all this *directly*—without the negative step!



Gives top quality at low cost . . . Another thing you'll like about this new paper is the fact that it's inexpensive to buy and use—produces intermediates as good as originals—and better—at low cost. Then, too, it is extra-thin, evenly translucent, assuring top-quality shop prints at practical, uniformly high printing speeds.

THE BIG NEW PLUS

Kodagraph Autopositive Paper

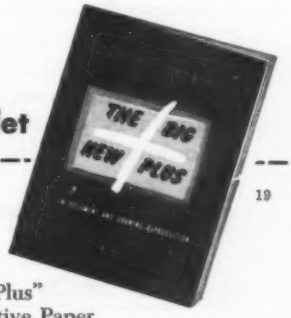
Designed for modern drawing reproduction . . . Kodagraph Autopositive Paper prints direct-to-positive in ordinary room light. Exposure is on familiar blueprint or direct process equipment, or in a vacuum frame. Processing is simple—photographic—without dark-room—without any time-consuming, detail-losing negative step. Finished Kodagraph Autopositive Paper intermediates are tough, long-lasting. They permit deletions—take drawing board changes readily—help reduce redrafting time and expense.

EASTMAN KODAK COMPANY
Industrial Photographic Division
Rochester 4, N. Y.

"Kodak" is a trade-mark

Comparison convinces . . . Have your reproduction department or local blueprinter make shop prints both from a Kodagraph Autopositive Paper intermediate and from the original. Then compare . . . you'll see why it pays to specify Kodagraph Autopositive Paper.

Mail coupon for *FREE* booklet



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Eastman Kodak Company
Industrial Photographic Division
Rochester 4, N. Y.

Please send me a copy of "The Big New Plus"
—your booklet about Kodagraph Autopositive Paper
and the other papers in the Kodagraph line. I have
☐ direct process ☐ blueprint ☐ contact printing equipment.

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Department_____
Company_____
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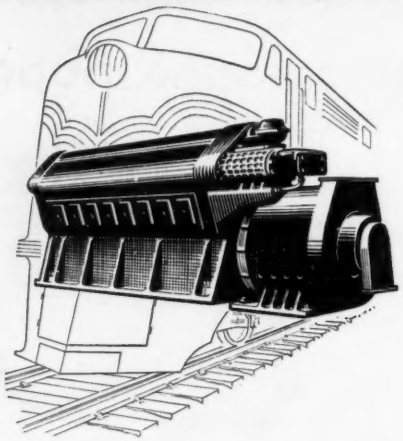
Kodak



dieselization

Diesel engines in railroad locomotives are piling up an enviable trouble-free performance record. Many castings in these engines contain specified Molybdenum contents.

When wear or heat must be resisted, and high strengths are required, the use of Molybdenum in cast iron is becoming more and more routine.

The listing of a cylinder liner for Diesel use in railroad locomotives containing 3.20% Carbon, 1.90% Silicon, 0.85% Manganese, with 0.20% Chromium, 0.40% Molybdenum and 0.45% Nickel added can be found on page 10 of our new booklet—"Applications of Molybdenum Cast Irons."

Dozens of other alloy cast iron compositions used in Diesel engines, as well as hundreds of other Molybdenum-containing compositions are similarly listed in this new booklet. Write for it!

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AUTOMATIC STAIRWAY LIFT. Handles materials from floor to floor. Adjustable to width, length, and pitch of stairway; can be used as lift over stairway or in open space. Pushbutton controls at each floor level. Complete with motor, cables, automatic limit switch and electric brakes. Moto-Flow Co., Bay City, Mich.

GASOLINE POWERED LIFT TRUCK. Four-cylinder, water-cooled, 26-hp "Cobra" engine. Hydraulically operated lift, tilt of load-carrying mast, and clutch. Welded body frame of heavy gage steel stampings. Turning radius, 61 in. 1000 or 2000-lb lift capacity, with 5, 7, and 9-ft lift heights. Transitier Truck Co., Portland, Ore.

Metalworking

DIE CASTING MACHINE. For zinc castings to 30 lb and aluminum castings to 10 lb. 800 tons locking pressure. Injection pressures to 40,000 psi. Die space between tie bars, 25 by 40 inches; die separation, 17½ inches. Hydraulically operated, electrical pushbutton controlled. Average of 3 to 4 zinc or 2 aluminum casting cycles per minute. Three models include gooseneck plunger type and cold chamber hand ladeling type. Automatic casting ejection. Kux Machine Co., Chicago.

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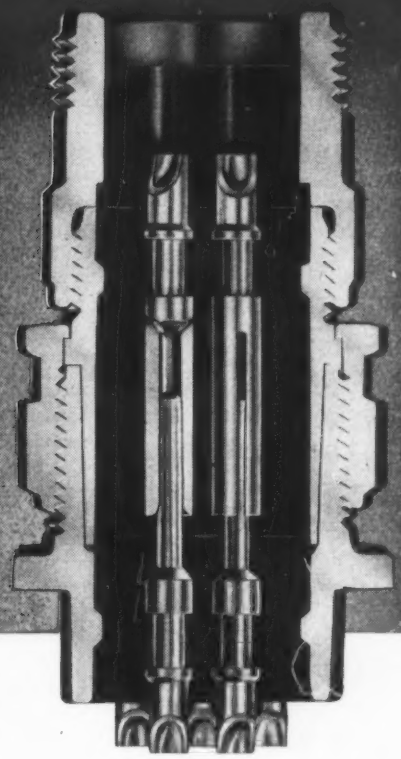

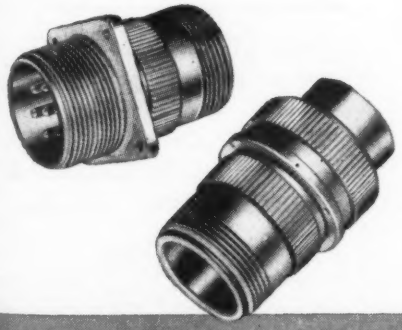

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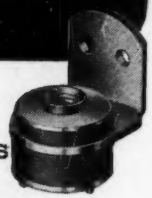

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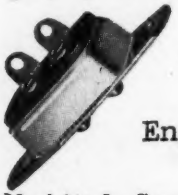

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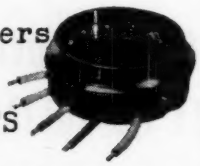

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BENCH GRINDER. For general tool room work. ¼-hp unit furnished with one coarse, one fine, 6 by ¾-in. wheel. Adjustable tool rest, wheel guards, transparent eye shields, and toggle type switch. Single-phase, 110-volt, 60-cycle motor. Torq Electric Corp., Bedford, Ohio.

DRILLING MACHINE. Belt drive from motor to four-speed geared transmission. Three optional spindle speed ranges. Head travel, 24 in.; 11-inch spindle feed. Table arm and table rotate on column. Feed mechanism driven from main driveshaft by steel worm and bronze worm gear. Sibley Machine & Foundry Corp., South Bend, Ind.

Testing

CREEP TESTING MACHINE. For short-time, creep-rupture tests at high temperatures. Motor-driven screw type; automatically holds constant loads to 100,000 psi on standard 0.505-in. diameter specimens at temperatures to 2200 F. Chart recorder automatically draws elongation versus time curve. Control circuit stops machine when specimen ruptures. Baldwin Locomotive Works, Philadelphia.

HARDNESS TESTER. For Rockwell testing metals and plastics. Cast aluminum body, frictionless spindle, positive tripping, and enclosed elevating screw. Three standard models having 8, 12 or 16-inch vertical capacities. Clark Instrument Inc., Dearborn, Mich.

VIBRATION TABLE. Handles 1000-lb specimens at 600 to 3600 cycles per min. Simple harmonic motion of table has total amplitude of 0.124-inch. Clear horizontal surface 4 by 6 ft will take equipment with center of gravity as high as 24 in. above table. Vibration produced by mechanical oscillator. Automatic controls stop main motor if vibration reaches 3700 cycles per min., if table vibrates laterally, if amplitude exceeds set amount, or if lubrication oil pressure fails. Baldwin Locomotive Works, Philadelphia.

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MACHINE DESIGN

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